LENNOX

ALL SEASON AIR CONDITIONING

(RESIDENTIAL APPLICATION)

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NOTE: All tables referred to in text appear in the back of the book.

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LENNOX ALL SEASON AIR CONDITIONING RESIDENTIAL APPLICATION

Section I, General

In the past we have been mainly concerned with Winter Air conditioning and the indoor comfort we create during cold weather by properly applying its functions of heating, air cleaning, humidification and circulation. Obtaining an equal degree of indoor comfort during the summer months is something most of us have recognized a need for and we have quite extensively accomplished it in places other than our own homes.

More and more people today are asking, "If we can enjoy this summer comfort in our offices, theaters, restaurants and stores, why must we tolerate summer discomfort in our homes?" The question is a well justified one and it is necessary that we, who specialize in home comfort answer this question with a practical means of obtaining all-season comfort in the home.

The new Lennox combination winter and summer air conditioning units adds controlled cooling, dehumidification, air cleaning and circulation during the summer months to the familiar winter air conditioning functions to give the homeowner complete, all-season comfort in his home.

Controlled cooling needs very little explanation. We know that during many summer days our homes are so uncomfortably warm that we would do almost anything to find relief. Controlled cooling keeps the temperature of the home at a comfortable level always, regardless of how hot the weather outdoors may be.

Unfortunately, high summer temperature is by no means the only factor contributing to discomfort in the summertime. High moisture content of the summer air, if not removed, may cause discomfort even though the indoor air is cooled. One of nature's methods of cooling the human body is by evaporation of perspiration. The faster this rate of evaporation takes place, the less discomfort is experienced in hot, humid weather. If the amount of moisture in the air in the home is too great there can be very little evaporation and nature's method of cooling is retarded, causing discomfort at temperatures that would otherwise be comfortable. Any summer air conditioning system that is entirely satisfactory must not only cool the interior of a home sufficiently but must also remove enough moisture from the air within it to maintain comfort.

Section II, Figuring the Cooling Load

The engineering data sheet for heat gain follows closely the procedures in the "Warm Air Heating & Winter Air Conditioning Manual" for figuring heat loss. The term Btu will continue to be used to evaluate building heat gain just as we have always used it as a unit of measuring heat loss; however, it will now be broken down into two groups, namely sensible and latent Btu.

Sensible Heat.

Sensible heat is the heat in a substance that can be measured with a dry bulb thermometer. A good example of purely sensible heat is the heat imparted to the room air by means of a furnace or boiler with no moisture added to the air for humidification. The heat felt when you touch a warm object is sensible heat.

Latent Heat.

Latent heat is the heat required to change the physical form of a substance without changing its temperature. If a block of ice whose normal temperature is 32 degrees is placed in a pan on the stove, heat will be added for a long while before all the ice finally melts, but the temperature of the ice and water mixture, if it is stirred constantly, will not change from 32° until after the ice has melted. The heat added is changing the physical form of the substance from a solid to a liquid and it is latent heat. Similarly, when water is boiled to form steam, or water vapor, 970 Btu of latent heat is added for every pound changed from water at 212° F. to steam at 212° F. This heat of the water vapor goes into and becomes a part of the total heat of the surrounding air. Water also evaporates at room temperature in proportion to the capacity of the air to hold additional moisture and this takes place under normal living temperature conditions. For example, for every pound of water evaporated at 75° there are 1,050 Btu transferred to the air as latent heat. This heat does not raise the dry bulb temperature of the room air.

Wet Bulb Temperature.

Unlike the winter heat loss calculation that uses only dry bulb temperatures for design conditions, summer air conditioning utilizes both dry and wet bulb temperatures. When a dry bulb thermometer with its bulb covered by a wick, wetted with water, is whirled rapidly through air containing water vapor, some water evaporates from the wick. The less water vapor already in the air, the faster the water will be absorbed, or evaporate from the wick. The more water vapor there is in the air, the slower the evaporation from the wick. Because heat is required to evaporate water and in this case the heat is being drawn from the atmosphere around the thermometer bulb, the temperature indicated by this wet bulb thermometer drops below the temperature read on an ordinary dry bulb thermometer. The lowest temperature finally reached by this wet bulb thermometer is the wet bulb temperature and the difference between this and the dry bulb thermometer reading is known as the wet bulb depression.

Wet and Dry Bulb Temperatures Relate "Sensible" to "Latent" Heat.

By knowing both the wet and dry bulb temperatures of the air one can determine the "Sensible"

and "Latent" heat content of this air either by calculation from published tables or by using one of the many psychrometric charts derived from these tables.

"Latent" Heat Also Contributes to Cooling Load.

It was explained why the degree of comfort experienced in the summertime is greatly influenced by the moisture content (the relation of "Sensible" to "Latent" heat) of the air. It follows that there must be some consistency in the ratio of the indoor dry and wet bulb temperatures if comfort is to be maintained. More specifically, as air is cooled it must frequently also be dehumidified, or have water vapor removed and additional refrigeration capacity must be provided to accomplish this.

In order to establish the total cooling load for a given space one must consider both the sensible and latent heat gains. Notice that the cooling load estimate sheet is broken down into two sections under the headings, "Sensible Heat Gains" and "Latent Heat Gains."

"Sensible" Heat Gains.

The section under "Sensible Heat Gains" is very similar to the Lennox Winter Air Conditioning heat loss sheet except that the effect of sun heat must now be added. This sun effect, or solar radiation, has never been considered in winter heat loss calculations not only because it helps when it does exist, but also the period of greatest heat loss usually occurs in the early hours of the morning when there is no sun effect. Conversely, with summer cooling, the period of greatest heat gain in a residence is about 4 o'clock in the afternoon when the heat from the sun is very considerable and it greatly increases the load imposed on the cooling system. Experience in residential work indicates that it is usually the greatest single contributing factor to cooling load.

"Latent" Heat Gains.

The latent heat gain in residential summer air conditioning is chiefly made up of the moisture brought in by outdoor air, either by infiltration or ventilation, and that produced by the occupants and their household activities.

To Size Equipment Add "Sensible" and "Latent" Heat Gains.

The two major divisions of the engineering data sheet are further broken down into subdivisions according to the method of calculation. We will discuss these one by one. In using these sections notice the similarity between this sheet and the heat loss calculation sheet. Careful use of this cooling load estimate sheet and the accompanying tables gives a total heat gain that is a reliable indication of the size of refrigeration package required to do the job. It must be kept in mind that the Total Cooling Load determined by the cooling load estimate sheet establishes the size of cooling equipment selected. This equipment will adapt itself to any load within its limits but it will fall short of handling a load that is beyond its capacity.

The cooling load estimate sheet provides for a refrigeration capacity that will cool the space figured by the amount of the design temperature difference selected. If the unit applied has less capacity than the load figured, the temperature maintained in the space under design conditions will be higher by at least an amount corresponding to the difference between the cooling load and the capacity of the equipment. For example, if the calculated load is 40,000 Btu based on a 20° indoor-outdoor temperature difference a three ton unit (36,000 Btu capacity) is applied, it would be 4,000 Btu or 10% short. This would result in at least a 10% or 2° warmer conditioned space than originally planned. "At least" is used because in areas where the latent load is appreciable, the conditioned space would be more than 2° warmer.

Be Careful of Oversizing.

As a general rule it is wise to avoid applying undersized equipment, but with summer air conditioning the use of excessively oversized equipment is also undesirable, even more so than with heating. It was mentioned that the summer air conditioning equipment removes latent as well as sensible heat from the space it conditions. The way it removes this latent heat is by condensing moisture on the cold surfaces of the cooling coil from the warm air passing through it. The only time the coil will condense this moisture from the air is when its temperature is substantially below that of the air, and these periods of low temperature occur only when the refrigeration compressor is running.

Owing to the fin construction of a coil it will hold a great deal of the water condensed from the air passing through it before any of this water begins to drain off. If the refrigeration compressor operates only a part of the time under design conditions, as with an oversized job, quite a substantial amount of this moisture which was condensed while the compressor was running will be held in suspension on the coil surfaces and re-evaporated back into the air being delivered to the conditioned space. This Re-evaporation, as it is called, can be so extensive as to result in practically no latent removal in extreme cases of oversizing the equipment. It is better to have equipment that runs most of the time, maintaining a cold coil so that this re-evaporation cannot take place.

The possibility of this Re-evaporation serves to emphasize further the extreme importance of using care in figuring the cooling load. It also indicates that sometimes it is wiser to select equipment that is slightly undersize rather than equipment that is greatly oversize. A great deal of judgment is required here in knowing just where to draw the line and we question that it is ever desirable to undersize more than 10%, even where the customer urges it.

A far better answer is to reduce the heat gain of the building by the addition of insulation or increased shading of exposed glass.

Give Directions on Construction Layout.

Notice that the reverse side of the estimate sheet is provided with ruled squares to facilitate making a sketch of the space to be conditioned. In addition to noting the usual dimensions and construction de-

tails, as with such a sketch for a heat loss estimate, one must, for the cooling load estimate, be sure to establish the direction the building faces together with any information on the type of shading used on windows. The direction that walls face and the shading of windows is extremely important in figuring a cooling load.

Selection of Design Temperatures.

In the upper, right-hand corner of the estimate sheet spaces are provided for entering the design conditions. Indoor design temperatures used in residential cooling work vary today anywhere from 75° to 80° with a tendency toward using the lower figures in regions of high latent load. The actual indoor temperature selected depends to a great extent upon an appraisal of the customer's requirements, but it is suggested that an indoor design no higher than 78° be used in localities where the outdoor design wet bulb is greater than 78°.

Outdoor design conditions for any locality may be obtained from such sources as the Guide of The American Society of Heating and Ventilating Engineers, from the local weather bureau, or from established practice in the particular locality.

Sun and Transmission Heat Gains.

The portion of the estimate sheet illustrated below provides for all the operations necessary to figure the total heat gain through outside walls, windows, doors and exposed roofs. This section, together with Tables I through IV, take into account not only the ordinary transmission gains but also the Solar Radiation gains and the manner in which thermal storage capacity of different walls affects these sun gains.

1. Sun and Transmission.

Direction of exposure				S	W					B. T. U. Gain
Gross wall areas				120	114					
Windows and doors	No.		Factor							
	1g		38	38						1444
	1g		178		12					2136
		T. D. table 3								
Net wall area	3e	20	2	82						164
	3e	28	3		102					306
Roof				Projected area						
(no attic)	236	50	14	180						2520
								Total		6570

The walls of the space must be classified according to the direction they face. Notice that six columns are provided in this table to allow for walls facing as many as six different directions. It will occasionally be necessary to use all six of these when figuring an entire house that is of odd-shaped construction.

For each gross wall area entered on the second line of this table, enter also the direction it faces in the space directly above. The example shown in the illustration assumes a room that is 15′ by 12′ with South and West exposed walls respectively, and with an 8′ to 11′ sloping ceiling. Notice that the letters "S" and "W" are entered in the spaces opposite "Direction of Exposure."

Window Heat Gains.

Table I gives the area of the more common windows in standard sizes. The glazing dimensions given on standard construction prints are the nominal size of one piece of glass only. For example a 24″ by 24″ double-hung window has a transmission area of 10.5 sq. ft., rather than the 4 sq. ft. one might assume.

For the illustration it is assumed that the South wall has a 16-light, double-hung sash with a glass dimension of 10″ by 12″. From the table this is an area of 16.5 or 17 sq. ft. Assume also that there is a standard 3′ by 7′ door in this South wall. This would be 21 more square feet to add to the 17 (doors are figured the same as windows for heat gain as well as heat loss), giving a total of 38 sq. ft., glass area for the South wall. Note where the figure 38 is entered in the illustration.

For the West wall assume a two-light, double-hung sash of 28″ by 24″ nominal dimension. From Table I this is an area of 12 sq. ft. Notice that the figure 12 in the illustration is entered one line lower. This allows space to enter the different transmission factor that exists for the West window.

In this instance, assume that both of these windows are single glass, equipped with inside venetian blinds. In Table II, at lg, under the column headed "S", read a heat gain of 38 Btu/hr per square foot for South glass with inside venetian blinds. Notice that lg and 38 appear in the illustration under their respective headings of "No." and "Factor". For the West glass the heat gain per square foot from Table II is 178 Btu/hr. This value and the key number are also entered in the section on windows and doors.

Wall Heat Gains.

Dropping down to the section titled "Net Wall Area" note in the columns for areas that the glass area has been deducted from the gross wall, and the net area of the West wall has been entered on the line below that for the South wall just as was done with the window areas.

Notice in addition to the "No." and "Factor" column, that another column, titled "T.D. From Table III" has been added. Table III gives an equivalent indoor-outdoor temperature difference to be used in figuring heat gain through different types of exposed walls and exposed roofs where there is no attic above. These temperature differentials in Table III take into account ordinary transmission, sun effect and thermal lag; everything necessary to obtain the total heat gain through the wall. In using this section of the estimate sheet always select a temperature difference for exposed walls or roofs from Table III; never use the design indoor-outdoor temperature difference.

If, for use in the illustration, it is assumed that the walls are of wood frame construction and are light

in color, we will select 20° from Table III as the total equivalent temperature difference for the South wall and 28° for the West wall.

Table IV gives the overall heat transmission for walls, partitions, etc., of various types of construction and at various temperature differences, such as are used for heat loss calculations. In the illustration it was assumed the walls are wood frame. If they are wood frame with 2" of blanket insulation, the heat gain factor for the South wall at 20° temperature difference will be 3 Btu/h per square foot from Table IV, and for the West wall at 28° it will be 4 Btu/h per square foot. The key number here is 3-e. Note how these values appear in the illustration.

Roof Heat Gains.

To demonstrate the use of a roof calculation it is assumed that the 12' by 15' room used for the illustration has a sloping shed roof with no attic space above. Here the sun effect is quite important, and for this reason calculation appears under this "Sun and Transmission" section. If there were attic above, ceiling loss would not be figured here, but in the next section under "Transmission Only." From Table IIIA the equivalent temperature difference for a 1" plain wood roof with 1" or 2" of insulation is 50° From Table IV the heat gain factor for this type of roof at a 50° temperature difference is 15 Btu/h per square foot. Note that it was elected to use 23-b rather than 23-c. In doing so it is assumed that the roof insulation in this example is 1" rigid material.

Notice that the area of this exposed roof enters in the space beneath the heading "Projected Area." The reason for the term "Projected Area" is that the actual area of peaked or sloping ceiling is rarely considered in figuring heat gain for summer air conditioning; instead use the projected area which is the same as a flat ceiling area if there were a ceiling. If one were to use the actual area of such a sloping or peaked ceiling the estimate would be too high because of the floor to ceiling temperature gradient. This gradient works to an advantage in summer cooling.

Total the Sun and Transmission Gains.

After tabulating all the factors and areas for the space it is a simple matter to multiply each area by its corresponding factor and enter the product of this multiplication on the corresponding line in the column headed "Btu Gain." In the illustration this has been done and the sum of these values appears as a total at the lower right-hand corner.

Transmission Heat Gains.

Surfaces such as Floors, ceilings beneath attics and partitions separating conditioned from unconditioned rooms are not directly affected by radiation from the sun and the lag, or heat storage effect, they have is so inconsequential as to be neglected in the ordinary heat gain calculation. For all practical purposes the heat gain through these surfaces results from transmission only and is directly proportional to the design temperature difference selected. In the illustration below and in the column headed "T.D." use this design temperature difference except for ceilings beneath attic space.

2. Transmission Only

	No.	TD	Factor	Area	B. T. U. Gain
Exposed Partitions					
Clg. under vented attic*					
Clg. under unvented attic†					
Clg under occupied space					
Floor over crawl space					
Floor over basement					
*Ceiling under vented attic: Add 15° to design temp. difference. †Ceiling under unvented attic: Add 40° to design temp. difference.				Total	
				Total	

Vented and Unvented Attics.

In the illustration notice that for a ventilated attic one should add 15° to the design temperature difference and in figuring ceiling under unventilated attic add 40° to the design temperature difference. This brings up the question as to just what constitutes a ventilated attic. By far the most practical and authoritative information to date on this contained in a publication by Tyler Stewart Rogers, Director of Technical Publications for Owens-Corning Fiberglass Corporation, and titled "Design of Insulated Buildings for Various Climates."

In brief, this manual recommends one square foot of free vent area for 100 square feet of insulated ceiling area in southern climates and one square foot of free vent area for every 150 square feet of insulated ceiling area in central and northern climates. If the area of the attic vents in the building falls short of these recommendations, add up to 40° to the design temperature difference in figuring ceiling loss. If the attic vent area is equal to or greater than these recommendations, add only 15°.

Partitions Between Conditioned and Unconditioned Space.

In commercial air conditioning it is common practice to select an intermediate design temperature difference in figuring the heat gain through exposed partitions, the reasoning being that the temperature of an unconditioned space adjacent to a conditioned one will not be quite as warm as outdoors. Naturally this same situation exists in residences, but its overall effect on the ordinary residential cooling load is so small that ordinarily it can be neglected. Use the design temperature difference to figure the loss through these exposed partitions in all but the most unusual cases.

Floors Over Crawl Space.

The design temperature difference should always be used in figuring floors over crawl space. Never count on an enclosed crawl space being cooler than outdoor temperature because of the cool, shaded earth below. If the crawl space is as adequately

ventilated as it ought to be, its temperature will be about the same as outdoor temperature.

Floors Over Basement and Slab Floors

A line for floors over a basement has been included but they are not usually figured in the heat gain calculation unless there is some unusual source of heat in the basement, or the basement is more freely ventilated than usual. There is no appreciable heat gain through slab floors on the ground and they are never figured in residential air conditioning. Therefore, they are not listed in the cooling load calculation.

Since the process of obtaining the Factor and construction type number from Table IV and multiplying this factor by the area is the same here as with the "Sun and Transmission" section, a specific example is not used to illustrate the use of this "Transmission Only" section.

Infiltration.

Since infiltration is directly related to the amount of crack opening that exists around windows and doors in the room and is in no way related to the cubical content of the room, the "Crackage" method of figuring infiltration rather than the "Cubical Content" method is recommended.

Determine Feet of Crackage.

Table V lists the running feet of crackage for windows of various sizes and types. Where two or more walls with windows surround the space do not figure infiltration for the total crackage because the wind obviously cannot blow from more than one direction at one time. The following rules should be used as a guide in figuring total feet of crackage:

1. For rooms with one wall exposed, figure the crackage for all the windows and doors in this wall.
2. For rooms with two walls exposed, take the wall in which there is the most crackage and use this figure as total crackage.
3. For rooms with three or more walls exposed, take one-half the total amount of crackage for the entire room.

Odd Windows.

Occasionally there are windows of a type not listed in Table V; in these cases measure the window and figure the feet of crackage. For double-hung windows the crackage is two times the height of the window sash plus three times the width. For double casement windows the crackage is three times the height of the window sash plus two times the width, and for single casement windows it is twice the height plus twice the width. The crackage for single doors is twice the height plus twice the width.

Figure CFM From Feet of Crackage.

Table VI gives the cubic feet per minute (cfm) of air that will leak in through every foot of crack during the summertime for windows and doors of various types of construction. Use this table only for summer air conditioning and never for winter heat load calculation. The rate of infiltration for summertime as given in this table is less than in wintertime because of lower prevailing summer wind velocities.

3. Infiltration.

	No	Running feet of crack	CFM Per Ft	Total CFM
Windows				
Doors				
			Total	

Rooms with two exposed walls: figure infiltration for wall with greatest crackage only.

Rooms with three or more exposed walls: figure infiltration for one half of total crackage

TOTAL: OTHER ROOMS
(Use only for room by room calculation)

In the illustration above, the total running feet of crack obtained from Table V is multiplied by the cfm obtained from Table VI to give the total infiltration. Doors are kept separate from windows in this calculation because the infiltration through doors is usually much greater than through windows.

Do Not Convert to Btu/h.

This total cfm of infiltration is not converted to Btu/h and added to the cooling load now. It is held for later comparison with the amount of outside air required for occupancy ventilation.

Outside Air Sensible Gain.

It was mentioned that the rate of infiltration in the summertime is not as great as in the wintertime. On many still summer days this rate of infiltration can drop so low that it does not adequately supply the normal ventilation requirements of the residence. In addition to this, it is quite generally accepted that more ventilation is desirable in a closed space in the summertime than in the wintertime.

Introduce Outside Air.

For these reasons we recommend that with every summer air conditioning installation provisions be made for introducing some outside air through a duct connected to the return air side of the system. The quantity of this outside air is based upon the number of occupants, and the cooling load estimate sheet suggests using 15 cfm of outside air per occupant. This recommendation is an average one that is recommended where light smoking is anticipated. It should seldom be necessary to figure a greater amount than this for residential work.

How Many Occupants?

The number of occupants to plan for in residential work depends to a great extent upon an appraisal of the customer's nature and habits. As a minimum, assume the normal number of persons in the family, plus any regular domestic help and add two as an

allowance for guests. This is not a hard and fast rule and it might occasionally be inadequate in cases where the lady of the house entertains frequently in the afternoon. Remember that 4 P.M. is the time of peak cooling load due to sun effect and thermal lag, and it is also a popular time for afternoon entertaining in the home for those who are in the habit of entertaining in the afternoon. In considering this problem, be particularly careful of larger homes because the number of occupants in these homes is subject to greater variation.

Compare Occupancy Ventilation with Infiltration.

The quantity of outside air for occupancy ventilation is determined by multiplying the number of occupants by the required ventilation air per occupant (15 cfm has been suggested for residential use). Now compare this quantity with the total infiltration for the entire house. If the entire house has been figured as a single space the total figure from Section 3, above, is the figure to use. It has been figured room-by-room, enter the quantity for each room in the table under "(A) Infiltration" and total them at the bottom of the table.

Select the Larger Quantity.

If the total infiltration is greater than the quantity of outside air required for occupancy ventilation, use it as the quantity of outside air to be introduced; if the occupancy ventilation air quantity is greater, select it. Having selected the proper air quantity (either (A) or (B) on the load sheet), multiply this by 1.1, times the design temperature difference to obtain the sensible heat load added by the outside air introduced.

Internal Sensible Gain.

The first and most obvious internal sensible gain and one that should always be figured is that resulting from the occupancy load. Every human being gives off a certain quantity of heat, part of it is sensible and about an equal part under most conditions of light activity is latent. In a residence, where the usual physical activity is considered very light, a value of 200 Btu/hr per person for sensible heat emission is a practical one to use. The sensible heat gain due to occupancy is obtained by multiplying the number of occupants by 200.

Unusual Internal Loads.

Also in Section 4 of the load sheet a space is provided for any other internal sensible heat gain of a special nature. At this time one of the major differences between residential and commercial air conditioning should be pointed out. In commercial air conditioning these internal loads due to appliances, etc., are of extreme importance because they usually occur at a time corresponding to the peak load resulting from outside conditions and, in many commercial establishments, these peak internal loads correspond to times when most customers are present and best performance of the system is desirable. This is quite different from the home where peak internal loads usually are early in the morning, again just before noon, and again about 5:30 to 7:00 in the evening. They correspond to times when family meals are being prepared.

It might justifiably be reasoned that the time of preparing the evening meal corresponds pretty closely to the 4:00 P.M. peak, and it does but, if the cooling capacity is augmented by the very considerable amount that would be necessary to take care of the heat given off by the range, the equipment would be seriously oversized at all other times.

Kitchen Exhaust Fan Recommended.

It is a far better plan to strongly recommend to the customer that he install a kitchen exhaust fan and leave the kitchen open to the rest of the house. A kitchen exhaust fan is an ideal residential exhaust system. It gathers the heated air and much of the water vapor from above the stove and removes it, at the same time removing much of the load that the stove would otherwise impose on the air conditioning unit.

Even where a kitchen exhaust fan is not to be installed do not add this range heat output to the cooling load estimate because of the danger of oversizing. It is better to explain to the customer that if the range is used very much on extremely hot afternoons the kitchen is apt to be too warm, and if the equipment were sized to accommodate this range load, the kitchen would be uncomfortably cool at all times except when the range is operating.

Return Air For the Kitchen.

If a kitchen exhaust fan is used, a return air opening in the kitchen is not necessary. If there is no kitchen exhaust fan, consider the kitchen just as any other room in providing for return air.

Other Internal Heat Sources

Television or radio sets account for only about 500 Btu/h at the most and in most cases can be ignored in the estimate. A laundry dryer, if in the conditioned space and not vented, cannot be practically taken care of with air conditioning. It must be vented and, if it is, there is no need to figure it. lighting is not figured because lights in a residence are not ordinarily turned on during the peak load at 4:00 in the afternoon.

If there is any unusual electrical load that must be figured, multiply the Watts input of this load as read from the appliance nameplate by 3.4 to get the sensible Btu/h to add to the estimate.

This completes the estimate of the sensible heat gains and they can now be totaled up in the right-hand column of the estimate sheet under "Room or Building Totals."

Latent Heat Gains.

It has been explained that latent heat is the heat of the water vapor in the air. If a good share of this latent heat is not removed from the outside air entering the conditioned space, the occupants will not be comfortable. To remove this latent heat from the entering air it is necessary, in most localities, to provide additional refrigeration capacity according to the latent heat content of the outside air as represented by the outside design wet and dry bulb temperatures.

This is the familiar air conditiong problem involv-

ing use of a psychrometric chart to determine the amount of latent heat removal necessary to satisfy a given set of design conditions. To save time in estimating, we have assumed an average set of satisfactory indoor design conditions and worked this psychrometric problem for a number of different outdoor design conditions to arrive at the table in the lower left-hand corner of the estimate sheet.

Latent Heat of Outside Air.

Knowing the outdoor design conditions determine from this table the Btu/h of latent heat to be removed for every cfm of outside air introduced. Multiplying this table value by the cfm to be introduced gives the latent heat gain added by this outside air.

Occupancy Latent Heat.

Another latent heat gain that must be considered in residential work is that of the occupants. It was mentioned that every occupant gives off about 200 Btu/h of sensible heat under conditions of light activity. Occupants also give off about an equal quantity of latent heat under the same conditions. The estimate sheet provides a space for multiplying the selected number of occupants by 200 to obtain the occupancy latent load.

Total Cooling Load Determines Equipment Size.

The Total Cooling Load is obtained by adding Total Latent Heat Gain to Total Sensible Heat Gain. The cooling equipment installed must have enough heat removal or cooling capacity to handle this Total Cooling Load. For example, if the Total Cooling Load turns out to be 36,000 Btu/h, apply a three-ton machine. A ton of refrigeration represents a heat removal capacity of 12,000 Btu/h.

Total Sensible Load Determines Air Quantity.

The total air quantity the machine should handle is not selected on the basis of this Total Cooling Load, but rather on the basis of Total Sensible Heat Gain. This is a little bit difficult to understand, and perhaps the easiest way to visualize it is to remember that the quantity of delivered air is determined by sensible load but the degree of coolness and dryness of this delivered air is determined by latent load. In carefully engineered commercial installations the characteristics of the cooling coil are selected on the basis of the ratio between sensible and total cooling load. This ratio actually evaluates the extent of the latent load.

To determine Total Air Quantity from the Total Sensible Heat Gain simply divide this Total Sensible Heat Gain by the quantity (1.8 x Td). Td is the difference between the temperature of the room or return air and the temperature of the air leaving the register. With the average residential installation a good figure to use for this temperature difference is 13°. Use it with the Lennox unit.

The Engineering Data Sheet can be used either to figure the whole house in one calculation or it can be used to figure the house room-by-room by using a sheet for each room. Where figuring room-by-room it is not ordinarily necessary to figure total air quantity but, instead, use the Branch Duct & Register Selection chart together with the trunk sizing method which will be given later. When the whole house is figured at once it is common practice to calculate the total air quantity and then divide it among the various rooms according to the square feet of floor area of each.

Room-By-Room Method More Accurate.

The room-by-room method gives a more accurate proportioning of the air and sizing of the branch pipes because these selections are based on the actual cooling load of the room rather than square feet of floor area. Floor area as an indication of cooling load can be quite deceiving if there is quite a difference in window area from room to room. With a job that is to be bid it is often practical to figure the whole house for making the bid then, if the job is secured, figure any critical areas room-by-room to be sure of using the correct branch sizes.

Section III—Air Distribution

If the cooling load has been figured room-by-room it is convenient to select and enter the branch duct and register sizes in the Branch Duct & Register Selection chart given on the Engineering Data Sheet and illustrated below.

Branch Duct and Register Selection.

Room											
Total Sensible Gain — Room Sensible											
Total Sensible Gain — O. A. Sensible											
Total Sensible Gain — Total B. T. U.											
Required Branch Area B. T. U.÷100											
Branch Pipe											
Riser Stack											
Supply Grille or Diffuser											
R. A. Register Free Area B. T. U.÷50											

Notice that this chart is divided into 10 columns to provide space for as much as a ten-room home. The three lines immediately below the room designation line and labeled Room Sensible, O. A. Sensible and Total Btu are used for adding and recording the total sensible heat gain for each room. In figuring a house room-by-room do not waste time figuring the outside air sensible gain separately for each room. Figure it for the entire house by totaling the individual room infiltration quantities and comparing this total with the occupancy ventilation load.

Branch Pipes and Ducts.

In this Branch Duct & Register Selection section the problem of dividing total outside air sensible gain among the various rooms presents itself. In most cases we suggest simply dividing it, roughly, according to floor area among the major rooms of the living area. For example, if the house has a living room and a recreation room, divide outside air sensible gain between these areas roughly according to floor space.

Occasionally some outside air sensible gain might be applied to a kitchen where a kitchen and dining area are combined, but never apply any of it to a kitchen area when exhaust is taken for the entire house through a kitchen exhaust fan. Ordinarily it is not advisable to apply any of this gain to the bedroom areas because their occupancy occurs at times other than peak load.

For Branch Area, Divide by 100.

After totaling sensible Btu/h for each room, determine the branch duct area required by simply dividing this total by 100.

Two Small Branches Often Better Than One Large One.

If this Required Branch Area is much greater than 50 square inches (the equivalent of an 8″ round pipe) it is generally advisable to plan on supplying the room with two or more branches rather than using a larger single branch. This is based on a consideration of the distribution of conditioned air within the room. Any room having a cooling load of 5000 or more Btu/h is a fairly large room and frequently needs to have air introduced at more than one place for proper distribution of the conditioned air.

For example if the required branch area is closer to 56 square inches than it is to 50 square inches give careful consideration to supplying that room with two 6″ runs rather than one 8″ run, particularly if the room is one whose performance requirements are critical. Air conditioning performance requirements are usually considered to be more critical in living areas than in bedroom areas.

One Large Branch More Desirable In Certain Cases.

One case where a large single run might be more desirable than two smaller ones is where a long, narrow room is supplied from one end. Here a single, large register gives better distribution throughout the entire length of the room. Another case where a large, single run might be considered is where ceiling diffusers are to be used and the room is nearly square. Here the choice of a large single diffuser is sometimes preferable to that of two or more smaller ones.

Use Standard Pipe and Fittings.

This design information contemplates that the three standard sizes of pipe, duct and fittings used in warm air heating will also be pretty generally applied to summer air conditioning. These are the 8″ round pipe with 14″ x 3¼″ rectangular duct and associated fittings, the 7″ round pipe with 12″ x 3¼″ duct and the 6″ round with 10″ x 3¼″ duct.

Where other than these standard pipe and duct sizes will be used figure the air quantity for each room; then size the entire system by the equal friction method selecting a pressure drop of 0.08″ to 0.10″ per 100 ft. Sizing other than 6″ 7″ and 8″ round pipes by simply dividing the total sensible load by 100, results in considerable error in pipe sizes and will cause balancing difficulties.

Registers and Ceiling Diffusers.

The selection of supply registers or ceiling diffusers for summer air conditioning involves somewhat more consideration than their selection for winter heating.

Cooling Requires Additional Consideration Of Air Delivery Problem.

In winter heating the capacity of a register or a diffuser is the main thing considered. A free area that is adequate to handle the necessary volume of heated air without excessive velocity and resulting noise is the major consideration. Since air velocity noise is of principal importance quite generous sizing is used so as to be sure that air velocity is substantially below the threshold of hearing. The shape of the air pattern, whether it is directed upward or downward and the distance it travels (its throw) although worthwhile considering, are of secondary importance in providing heating comfort.

The reason behind this is that in winter heating air is discharged that is warmer than the air within the room. From a high wall register or a ceiling diffuser it tends to rise above the zone of occupancy because it is warmer and, hence lighter than room air. Any draft that might be felt in its path of discharge is above the heads of the occupants and rising.

In cooling just the opposite situation exists. Here air that is cooler than room air is discharged above the heads of the occupants. Instead of rising out of the occupancy zone it drops very quickly down into this area and creates a serious draft problem unless it is well mixed with room air and slowed to less than about 50 feet per minute before it reaches this occupied zone.

For this reason the pattern shape, upward or downward deflection and throw of air from the ceiling diffuser or wall register all assume more importance in summer air conditioning.

Select Registers On Manufacturer's Performance Data.

The makers of summer air conditioning registers and ceiling diffusers provide tables for selection of their equipment on the basis of the following four considerations:

a. Capacity, as limited by noise.

b. Pattern shape.

c. Degree of upward or downward deflection.

d. Throw distance.

The methods of selection of registers and ceiling diffusers from the manufacturer's tables are carefully described in their instructions accompanying these tables and these manufacturer's tables and instructions or their local representatives must be consulted in selecting registers or ceiling diffusers.

Use Quadrant Dampers In Branches.

With summer air conditioning systems it is always necessary to use a quadrant damper in the branch duct to each supply register or ceiling outlet, even though the register or outlet used has its own volume

damper. Perhaps there is some question why both the quadrant and volume dampers are required for summer air conditioning whereas volume dampers only have always been entirely satisfactory for the straight warm air heating job. The reason for this is that, in balancing the job, if air quantity is reduced at a register by closing down its volume damper only, this volume damper must be closed quite a long way before any appreciable reduction in volume occurs. A reduction in opening area at the register without appreciable change in air volume results in quite a substantial increase in throw distance. This can easily result in high velocity air traveling clear across the room, striking the opposite wall and rebounding down on the occupants. On the other hand, if an attempt is made to increase air quantity by opening a volume damper at the register face, the tendency is to reduce the throw distance causing the cool air to drop into the occupied zone before it has mixed with the room air. This causes complaints of drafty conditions also.

Adjust Air Quantity With Quadrants and Throw With Volume Dampers.

Where quadrant dampers are installed adjust the air quantity with them and then proceed to eliminate drafts by adjusting the throw of each register with its volume damper. The recommended throw of a wall register is three-fourths of the distance across the room. This is a good figure to remember in selecting registers from a manufacturer's tables, but it is not very practical to measure it in the field. After adjusting the air quantity it is best to simply adjust the volume damper behind the register until drafts are corrected. Do not waste time measuring the exact distance registers are throwing. The radius of throw from ceiling diffusers should also reach about three-fourths of the way from the diffuser to the wall.

Return Air Registers and Grilles.

No special provisions apply to sizing and locating return air registers and grilles for summer air conditioning that do not already exist for warm air heating. No particular advantage in cooling is to be gained by locating return air grilles beneath windows; however, air distribution is improved by locating them in walls opposite the supply registers. The important consideration with return air grilles in residences is to be sure of having adequate free area in order to keep the face velocities low and thus eliminate a source of drafts. The branch Duct & Register Selection chart suggests determining the free area by dividing the Btu/h each grille will handle by 50. This results in a face velocity of about 300 feet per minute which is recommended for low-wall returns in residences. Remember that the free area of a grille is only about 70% to 80% of the nominal area. Safest practice is to divide the free area by 0.75 to get the actual area of the grille you will need. Volume dampers are not ordinarily required for return air openings in residential air conditioning.

Return Air Branches.

Return air branches can be sized on the basis of 100 Btu/h per square inch of branch area, just as supply branches, or they can be sized by the equal friction method. If the Btu/h the branch is to handle is divided by 100 to determine branch size, it is all right to exceed 5000 Btu/h (50 square inches of area) per branch with return branches, but never make a return smaller than 28 square inches or 6" round using this method.

Outside Air Intakes and Exhausts.

The required quantity of outside air should be positively introduced into the system by means of an outside air intake duct connected to the return side of the system. It has been mentioned that the kitchen is a good place to exhaust some of it. Another desirable location for some exhaust is from the bathroom; however, a natural rather than a power vent is usually satisfactory here. If a kitchen power vent is used for the exhaust, be certain there is some means of adjusting the quantity of air it handles. This air quantity may have to be reduced to prevent over-venting.

Sizing the Intake.

The outside air intake duct connected to the return side of the system should be sized on the equal friction chart for about 0.08" per 100 feet, or according to the table below:

Total cfm Outside Air	Pipe Sizes
100	6" round or 10 x 3¼
150	7" round or 12 x 3¼
200	8" round or 14 x 3¼
250	9" round or 12 x 6

This outside air duct should be provided with a manual damper that is reasonably accessible and its outdoor opening should be screened and hooded for protection from the elements.

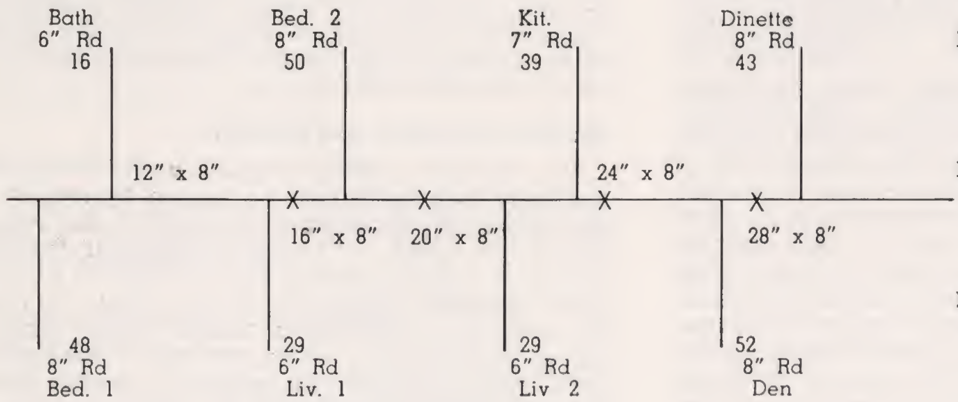
Trunk Sizing.

If the equal friction method is used in sizing branches it should be continued in sizing the truck ducts. If branches are sized by dividing sensible Btu/h by 100 as shown in the Branch Duct & Register Selection chart, then size the trunk as explained below.

"Modified Extended Plenum" or "10% Reduction" Method.

A simplified system recommended is the "modified extended plenum" or "10% reduction method" whose use is common practice in warm air heating. The smallest cross section at the end of this trunk is 12" x 8" which may seem somewhat oversized for the end branches but the advantage of a slight static regain is present in this type of trunk and this assists materially in feeding these end branches. Branches may be taken from the trunk on any of the four sides, but standard "takeoff" fittings should always be used.

Referring to the illustration, page 10, note that a line drawing has been made of a representative air conditioning trunk and its branches. The explanation of this method is as follows:



Bed. 1	48	
Bath	16	
Liv. Rm. 1	29	
	93	12″ x 8″
Bed. 2	50	
	143	
—10%	14	
	129	16″ x 8″
Liv. Rm 2	29	
	158	
—10%	16	
	142	20″ x ″8
Kitchen	39	
	181	
—10%	18	
	163	20″ x 8″
Den	52	
	215	
—10%	22	
	193	24″ x 8″
Dinette	43	
	236	
—10%	24	
	212	28″ x 8″

1. As mentioned before make a simple line sketch of the trunk with the various branches leading off in their correct relative positions.

2. At each branch line record the "Required Branch Area" as taken from the "Branch Duct & Register Selection" chart. Notice that this is "Required Area" and not the area of the branch pipe used. This is important. There is no need to put any more area in the trunk than the rooms need, even though branches might be oversized because of using commercial sizes.

3. Now start at the far end of the trunk and begin adding to see how many branches come within the 96 square inches available in that last, smallest section of trunk which is 12″ x 8″.

4. At this point in the trunk, after taking as many branches as come within the 96 square inches capacity, use a transition fitting that takes the trunk up to 16″ x 8″ or 128 square inches.

5. At this point add the next branch and then reduce the total by 10%. Now continue to add branches and reduce the total by 10% each time one is added until the reduced total reaches the 128 square inches area of the 16″ x 8″ section. (Make no 10% reduction in the total after adding any branch that is smaller than 15 square inches.)

6. When it becomes necessary to expand the trunk further, go up another 4″ in width to a 20″ x 8″ etc. Always increase the trunk size in 4″ increments.

7. Follow this 10% reduction scheme every time a branch is taken off until the main trunk has been reduced no smaller than 70% of all required areas that the trunk must feed. If it should do so by any appreciable amount, size the trunk instead by the equal friction method using a pressure drop of 0.08″ per 100 ft. of duct. With the equal friction method it is still alright to reduce by increments of 4″ in width if doing so reduces installation costs.

Turning Vanes.

Turning vanes must be used in all supply and return trunk elbows unless there is room to make an elbow whose inside throat radius is equal to the width of the duct. It is also recommended that turning vanes be used in stackheads behind wall supply registers. These improve the register performance to a marked degree and greatly reduce the possibility of noise at the registers. If it is necessary to install a register directly in the side of a trunk duct place turning vanes in the duct behind it at an angle as illustrated below. These vanes should be pivoted on rivets top and bottom so that they can be closed or opened when balancing the system.

Section IV—Duct Insulation.

Even where power rates are most inexpensive and fuel rates are relatively high the cost of operating air conditioning is more than twice as much as the cost of heating per Btuh handled. This means that well-insulated ductwork must be used in order to keep the waste of cooling capacity to an absolulte minimum. Skimping on insulation is one of the poorest possible ways of cutting the cost of an air conditioning job. Not only does it give trouble with balancing the system but also it leads to constant complaints from the customer on the cost of operation no matter how well the system works for him.

Supply Duct Insulation.

All supply ductwork that is run through attics should have at least 2″ of insulation. Supply ductwork running through unconditioned spaces should have at least 1″ of insulation. Where a supply duct is to be furred into the ceiling of a conditioned space and the space above is unconditioned or if the con-

ditioned space is a kitchen and the space above is conditioned this ductwork should have 1" of insulation.

Supply ductwork that is run exposed in the conditioned space does not ordinarily need to be insulated unless the space in question is a kitchen. Riser stacks in partitions with conditioned space on both sides do not need to be insulated but if the opening cut to receive them into the stud space leads into unconditioned space a careful seal must be made between the sides of the opening and the duct. This stud space must also be carefully sealed at the bottom if it has any way of communicating with unconditioned space.

Return Duct Insulation.

Return ductwork running through attic space should have 1" of insulation. Return ducts through unconditioned space ordinarily need not be insulated. Return ducts in walls and partitions whether surrounded by conditioned space or not require no insulation unless the design wet bulb is higher than 80°, and then the opening to receive them into the partition space should be sealed the same as with supply stacks dropping into partition spaces. A good material for these air seals on supply and re-return risers is Bauer & Black Fiberglas Tape or an equal fire resistant tape. Asbestos tape may be used but as a general rule it is not very satisfactory for use on air conditioning ductwork. The tendency toward presence of moisture on air conditioning ducts and the moisture retaining properties of asbestos combine to set up an unsightly and sometimes odorous mold in the glue used to apply it.

Vapor Barrier.

All insulation applied to the exterior of air conditioning ducts must be provided with a vapor barrier completely surrounding the outside of this insulation or the insulation itself must possess vapor barrier properties. In addition to this the insulation used ought to be of a type that is not affected by moisture. Many insulating materials today can be obtained with a vapor barrier material already cemented to one side of them. Since this eliminates the separate operation of wrapping the vapor barrier after insulating, they are more convenient to use. This insulation with vapor barrier should always be applied with the vapor barrier out or toward the warm side.

Inside Insulation.

Some installers prefer to apply insulation to the inside of ductwork because it has excellent sound absorbing properties and there is no worry about a vapor barrier because the metal duct serves as the vapor barrier. Where interior insulation is applied always use a type specifically recommended by the manufacturer for this type of application; otherwise the system is apt to blow shreds of insulation from the registers to the extent that it will be a continual nuisance to the homeowner.

Condensation Must Be Prevented.

Aside from loss of cooling capacity, another reason why there is concern about proper application of insulation and vapor barrier material is to prevent condensation of moisture on the ductwork. Nearly every air conditioning contractor doing business in humid climates has at one time or another suffered the misfortune of having to do some extensive redecorating in the customer's home because of some oversight in the application of insulation and vapor barrier to overhead ductwork. If any moisture condensation on attic ductwork is not discovered and remedied quickly it can ruin the ceiling beneath it and the misfortune is that sometimes this dripping of condensation is not discovered until serious damage has been done.

In climates where the design wet bulb temperature is above 78° this condensation problem is so serious that every square inch of supply duct exposed to unconditioned air must be treated with insulation and vapor barrier. In places where obstructions such as ceiling joists preclude an exterior application of this material, an interior application must be made in the spots that cannot be reached from the outside. In humid climates do not leave a single square inch of attic supply ductwork uninsulated.

Insulating For Sound Absorption.

It was mentioned that some air conditioning contractors prefer to apply insulation to the interior of ducts in order to eliminate the vapor barrier and absorb sound. This application of interior insulation for sound absorption has a great deal of merit in residential air conditioning; so much, in fact, that many contractors purposely line the interior of both their supply and return trunks for the first 10 feet, or so, either way from the unit. They do this even though they may finish the rest of the job with exterior insulation and vapor barrier. This practice is recommended even to the extent of adding some extra length to the return trunk, where necessary, in order to install about 10 feet of this treatment.

Oversize For Interior Insulation.

Where interior insulation is applied to a trunk duct increase the cross section of the trunk to the extent that the finished opening with insulation in place is no smaller than the calculated area of the trunk. If it is applied further, to the extent of using it throughout the entire trunk and branches also, oversize even further to allow for the greater surface resistance factor of the insulation. Information on the extent of this oversizing necessary can be furnished by the supplier of the insulation material.

Flexible Duct Connectors.

Another practice, for reducing sound level, that is recommended wherever space permits its use is the application of flexible duct connectors. These should be made of heavy asbestos cloth, secured to the air openings of the unit and the trunk ducts with "S" strips and metal screws rather than asbestos tape or glue. These flexible connectors need be no wider than about 4" and they should be located at the connection of the trunk duct to the unit.

Section V—Adapting Summer Air Conditioning To Existing Warm Air Heating Systems.

In applying summer air conditioning to the existing warm air heating system the major problem is with the air distribution system and what revisions it needs to adapt it to the delivery of the larger quantities of refrigerated air.

Make a Sketch of Heating Layout.

When surveying the job for the cooling load estimate make a complete sketch of the heating layout showing all the supply and return branch and trunk sizes, as well as register and grille sizes. Also indicate on this sketch all elbows and bends in the ductwork, particularly those in the trunk ducts and whether or not these trunk elbows have turning vanes if it is possible to tell. While examining the ductwork check whether quadrant dampers are installed and operating and how accessible the ductwork is for insulating, noting on the sketch any ducts that will have to be replaced because they are not accessible to insulate.

Examine Blower Carefully.

If there is a chance of using the blower now installed with the heating system measure the diameter of its wheel and the width of its scroll. It cannot be predicted accurately just by these measurements that this blower can be speeded up to handle the increased air load of cooling without causing noise trouble, but measuring the blower now saves wasting time with blowers that will not work.

As a general guide based on the experience of those who have applied air conditioning to heating systems an existing furnace blower with a wheel diameter smaller than 12″ will not be satisfactory for quiet operation with any cooling load of three tons or more. The chances are very good that any good, standard 12″ blower will handle a three-ton system. The chances are fair that a 12″ blower will handle a five-ton system if the width of its scroll is at least 15″ and the resistance of the system is moderate. It is not advisable to waste time considering applying more than five tons of air conditioning with a 12″ blower on the usual residential conversion job. At least a 15″ blower with an 18″ scroll width is needed for these larger conversions.

Check Motor and Drive.

If the size of the blower itself appears adequate examine its drive carefully, noting the pulley diameters, the condition of pulleys and belt and the setting of the variable pitch motor pulley. If the motor pulley is not of the variable pitch type or if it is set for maximum speed now, figure replacing the complete drive.

Next check the blower motor nameplate. Read the horsepower of the motor and the current it draws. At least a $\frac{1}{4}$ horsepower motor will be needed for two tons of air conditioning, $\frac{1}{3}$ or $\frac{1}{2}$ for three tons and $\frac{1}{2}$ or $\frac{3}{4}$ horsepower for five tons. Run the blower and if possible check the actual current drawn by the blower motor. If it is now drawing almost its nameplate current it will be overloaded when cooling is added. Plan to replace it with the next size larger motor.

Run the Blower.

While the blower is running listen for any excessive or unusual noises. If the blower is noisy now it will probably be more so when driven at a higher speed. Unless it is obvious how this noise can be corrected figure on replacing the entire blower even though it appears to be suitable otherwise.

Figure Cooling Load and Required Branch Areas.

Now figure the cooling load and the required branch pipe area for each room. Also figure the approximate cfm required for each room by dividing room sensible load by 24. Compare the required branch pipe areas with the actual, installed branch areas on the heating layout. There will be some that are adequate in area and some that are not.

Those branches that are adequate by this comparison will handle their cooling load without trouble if they are well installed, have a quadrant damper and can be insulated. For those branches that are not of adequate area there are three choices; replace them with a branch that is larger, supplement their capacity by adding another branch to the same room, or force them by raising the static pressure of the entire system and closing down the quadrant dampers of those branches whose areas were adequate.

Check Questionable Branches With Equal Friction Chart.

Use particular care in this latter choice because it is so difficult to draw a definite line as to just how much smaller than required the branch can be and still work. It depends mainly upon the reserve capacity in the blower. It is dangerous to assume that an undersized branch will work on this job simply because one worked under similar circumstances in the past because of the vast differences in systems and blowers. A much safer guide is, with the required cfm for the branch in question, determine the required round pipe size from an equal friction chart at 0.10″ per 100 ft. If this required area is still more than the area available, it is best to replace the branch. If the system has a pressure drop much above 0.10″ per 100 equivalent feet of pipe there will be difficulty in obtaining quiet operation with the average furnace fan, particularly if the pipe runs are very long.

There are ways of compensating for these undersized branches such as providing them with elaborate vaning at the elbows and special, enlarged takeoff collars, or by lowering the cfm of the entire system, providing more refrigeration capacity and delivering cooler air, but these expedients also increase the cost of installation and frequently the operating cost, too. Furthermore, they endanger the performance of the entire system unless they are handled by someone well acquainted with air conditioning design through years of experience.

Check the Trunks Also.

In checking the adequacy of the trunk size time is saved by using the equal friction chart together with the chart of equivalent rectangular duct sizes. With the 10% reduction method sizing began with the small end of the trunk and worked back toward the unit. With the equal friction method sizing begins at the unit and progresses to the small end as follows:

1—Select the pressure drop to be used as a basis of sizing. In this case 0.10" per 100 ft. is recommended.

2—Now, opposite the total cfm the trunk will handle, and on the vertical line at 0.10", read from the diagonal line the round pipe size needed.

3—Knowing the width of the existing trunk duct from the layout enter this as one side of a rectangular duct in the chart of equivalent duct sizes. Follow across horizontally until the curved line for the round duct size found in the equal friction chart is intersected. At this point drop vertically and read the necessary depth for the trunk.

4—If the first section of trunk on the heating layout is not this deep, plan on replacing it.

5—Now deduct the cfm carried by the first branch and with this new total cfm, repeat steps 2, 3 and 4.

6—Continue to do this until the last, smallest section of trunk is reached. This last section of trunk should be 12" x 8" and the total cfm it handles should not call for greater than a 10.5" round pipe at 0.10" on the equal friction chart.

Existing Supply Registers and Diffusers.

Before now nothing has been said about supply registers and their location, and these considerations are important. Sidewall registers must be above head height (at least 6½ feet above the floor) and situated so that the air leaving them will be well distributed throughout the room. It is hard to make a register located in a corner do a very good job of distributing the cooling throughout a large room, unless the corner happens to be next to a long, outside wall. In addition to this, registers for cooling should have a volume damper arrangement for reducing their face area without deflecting the air stream up or down and they should have means for adjusting their discharge pattern from side to side.

If the existing registers do not appear satisfactory on the basis of the above considerations or if there is evidence of considerable leakage around them, figure on replacing them. If there are ceiling diffusers and no accurate information is available on their characteristics, plan to replace them also.

Turning Vanes.

It has been mentioned that turning vanes in riser stacks behind wall registers are desirable on new installations. They are even more so on conversions where most of the stack velocities will be higher.

Examine Return System.

The sizes of existing return air grilles should be checked by dividing the sensible cooling load each will handle by 50. If they are undersized, replace them with larger ones or add more return grilles and branches. The size of return branches and trunks should be checked on the same basis recommended for the supply system.

Seasonal Unbalance.

The problem of seasonal unbalance, though it does exist, does not cause as much trouble as many would expect. Rooms with large south and west glass exposure in southern climates tend to overheat in the winter if balanced for summer air conditioning; conversely, rooms with large north glass exposure in northern climates tend to give underheating trouble in the winter when balanced on the summer air conditioning cycle.

A way to partially overcome this heating override in southern climates is to break the air quantity supplied to these south and west rooms up among more, smaller branches and registers and to instruct the homeowner to close off certain ones in the wintertime. For north rooms in northern climates break up the air quantity among more, smaller branches and registers and add one more than required, with the intention that it can be opened in the wintertime and closed in the summer.

Check the Availability of Services.

Services to the unit need to be checked either for an addition of air conditioning to an existing system or an installation of a complete new system. A water supply, drainage and an adequate electrical service must be available at the equipment location.

Where to Locate the Unit.

When applying air conditioning to an existing heating system, locate the condensing unit anywhere that is convenient for making direct connections into the discharge side of the heating and air handling unit. One exception to this is an attic location. The average residence is not built to support the weight of heavy, moving machinery in the attic. Such a location invites serious noise trouble with even the quietest air conditioning equipment. In addition to this, high ambient temperatures existing in most attics cause overheating of refrigeration equipment.

A crawl space location for the cooling equipment is acceptable provided it is dry and headroom and service access are sufficient.

Do Not Place Cooling Coil in Return Air.

It was mentioned that the condensing unit should be arranged for convenient duct connection to the discharge side or warm air plenum of the furnace. The reason for this suggestion is that it is not good practice to locate a cooling coil on the return air side of a warm air furnace. The inside of the furnace heat exchanger is exposed to unconditioned air during the summer months and, with cold air passing over the outside of this heat exchanger, the tendency is for condensation to form on the inside causing rust and shortening the life of the furnace.

Bypass Arrangement Is Best

The best possible arrangement is a complete bypass (alternate paths for the air in summer and winter) with seasonal changeover dampers such as employed in the Lennox Combination Unit. This not only eliminates any possibility of condensation troubles but it also serves to reduce the total static pressure of the system.

Section VI—Condenser Cooling Methods.

Contrary to widespread belief, the refrigeration equipment does not itself remove the heat from the conditioned space. The cooling system for the condenser carries away all the heat removed from the conditioned space plus the heat resulting from the work done by the compressor. The refrigeration machine is a device for raising the temperature level, or intensity of the heat from the conditioned space so that it can be carried away by the outdoor air or by water. For this reason, an understanding of air conditioning includes an understanding of the methods used to remove the heat from the refrigeration machine.

Air Cooled Condenser.

The first and most familiar method is the air-cooled condenser. An air-cooled condenser is very similar to the evaporator, or cooling coil in an air conditioner. It relies on the passage of outside air over finned tubing to remove the heat from the refrigerant; in larger sizes it requires a blower or fan to force enough air over its surfaces to obtain sufficient cooling. Air-cooled condensers for cooling equipment much larger than 1 Hp. are very large in size and wasteful of copper; and they have the added drawback for air conditioning of performing least well on hot days when their best performance is needed most.

Water Cooled Condenser Wasting Water.

Another familiar system uses finned coil surface in a sealed container with cold water for cooling the refrigerant circulating either through the coil or through the shell surrounding it. The water is drawn from the city main and after serving to cool the condenser it is wasted down the drain. This system is extremely satisfactory in places where there is an abundant supply of inexpensive cold water but there are not very many such places and the number of them is becoming fewer and fewer. This method of cooling wastes at least one gallon of water per minute per ton of cooling even in localities where water temperatures are nearly cool enough to permit using the water directly through a cooling coil for air conditioning without any refrigeration equipment.

Evaporative Condenser

The evaporative condenser is another method of cooling with water that vastly reduces the amount of water used. In this device the warm refrigerant is circulated through either plain or finned tubing much like the air cooled condenser, but water is sprayed over the outside of the coil and permitted to evaporate. This rate of evaporation is generally speeded up by use of a blower. The evaporative condenser is an efficient cooling device and very conserving in its use of water since it relies upon the evaporation of the water as a source of cooling. It's principal shortcoming is its tendency to collect lime on the condenser tubes. This just about rules it out in localities where the lime content of the water supply is high.

Air-Water Condenser.

Still another variation is the air-water condenser; this is almost identical to the evaporative condenser except that it relies on air alone when loads are light but sprays the condenser coil with water when the load becomes heavier. This device is subject to the same shortcomings as the evaporative condenser so far as lime is concerned.

Spray Pond.

Another distinct cooling method is the spray pond. Here a condenser like that used with the waste-water system is employed, but the water, rather than being wasted, is pumped through a group of nozzles, much like lawn sprinklers, arranged in a shallow pond. The purpose of the nozzles is to break the water into a fine spray so as to mix it well with the air and obtain a high rate of evaporation. This evaporation cools the remaining water almost to the outdoor wet bulb temperature, and in the process the latent heat of evaporation is carried away by the air. The unevaporated, cooled water is collected in a pond beneath the nozzles and recirculated through the condensor. A city water connection provided with a float valve and known as the "Makeup" line is used at the pond to add fresh water to replace that which evaporates. The operating principle of this system is quite sound but the pond requires a great deal of space and if very much of a breeze is blowing the mist carried away from the pond is a severe nuisance as well as a waste of water.

Cooling Towers.

The final method of cooling, and the one that will be discussed to some length is the cooling tower. There are three distinct classes of cooling towers and each of them will be treated separately.

Natural Draft Tower.

The natural draft cooling tower is essentially an improved spray pond. It uses spray nozzles as the principal means of breaking up the water to secure rapid evaporation. Unlike the spray pond the natural draft tower has its nozzles situated at the top and directed downward. Surrounding the spray of the nozzles is an enclosure of wooden louvres designed to permit free passage of air but to minimize the escape of water in the form of mist. The wetted surfaces of these louvre boards also serve to increase the rate of evaporation. A wooden pan, or sump at the bottom collects the cooled water to be pumped back to the condenser.

The universally accepted material for construction of natural draft towers is Redwood. This wood

is used quite generally in the construction of all types of water cooling equipment because of its excellent resistence to rot. The natural draft tower requires a lot less space than a spray pond but it is still a large and rather unsightly piece of equipment and it fails to eliminate drift to the extent that it is not recommended in localities where the prevailing wind velocity exceeds 4 miles per hour. Lawsuits over the nuisance created by the drift from natural draft cooling towers are frequent occurences in areas where wind velocities are higher.

To conceal the appearance of natural draft towers homeowners frequently make a planting of shrubbery around them or urge that they be placed behind a garage. Such attempts to conceal these towers usually impair their performance seriously by reducing the movement of air through them.

Performance of Natural Draft Towers.

In spite of their disadvantages natural draft towers are extensively used in residential work because of their low first cost as well as low maintenance cost. At first thought their operating cost may appear lower but this is a little misleading because their performance does not equal that of other types of towers. The performance of a cooling tower is based upon the gpm of water cooled through a definite range for a specified approach to wet bulb temperature. This "Range" of a cooling tower is the difference between entering and leaving water temperatures. The "Approach" is the difference between leaving water temperature and design wet bulb temperature. For the same quantity of water circulated a natural draft tower will not, as a rule, cool through as great a range or approach the wet bulb temperature as closely as will forced or induced towers. This usually makes it necessary to pump a little more water to carry away the same amount of heat. Another consideration is that natural draft towers often require a greater pumping head, or pressure than other types of towers.

Induced Draft Towers.

Another type of cooling tower that is frequently used is the induced draft tower. This tower makes use of a fan to draw air through the falling water droplets and by virtue of using this fan it can be much smaller and more compact. Instead of using spray nozzles to break up the water, induced draft towers generally employ what is called a "Pack", or "Deck Fill". This is a compact assembly of redwood pieces placed in such a manner that they break the falling water into small droplets and also permit the passage of air through the tower.

The performance of induced draft towers is the best of the three types and they are so compact that making them inconspicious does not present much of a problem. The fact that they have a motor-driven fan makes them more expensive to buy and install but they are generally better suited to residential work than are natural draft towers.

About the only disadvantages to induced draft towers are their noise of operation and their comparatively high maintenance cost. Because their operating sound level is characteristically higher than other types of towers they should be located as far from the house as practical and by no means near sleeping quarters or neighboring dwellings. Their high maintenance cost occurs principally in coastal regions where there is salt in the atmosphere, or where other corrosive chemicals may be present in the air. The saturated air off the tower, passing through the fan, combines with a corrosive atmosphere to greatly shorten the life of the fan.

Forced Draft Towers.

The third common class of cooling tower is the forced draft tower. It is very similar to the induced draft tower except that instead of using a fan to draw air through the wetted "Pack" it employs a blower to push the air through. The performance is a trifle below that of the induced draft tower because an even distribution of air is harder to attain but the difference is small. They are generally quieter in their operation where they use a squirrel-cage type blower rather than an axial fan. Their principal disadvantage is that their blower is generally arranged with a belt drive and is exposed. In residential use this blower must be enclosed and protected by screen in some manner to prevent injury to children.

In some of the Gulf Coast areas, where the salt content of the air is high and the water is also corrosive, a forced draft, redwood tower is about the only type of tower that will last for any reasonable length of time.

Tower Selection.

For the Lennox, three ton cooling equipment to develop its full capacity no higher than 95° water leaving the condenser, or to the tower is recommended. In areas where the design wet bulb is 80° this means that there is only a 15° spread between entering water and wet bulb temperatures. The usual conditions for operating a tower are where the range approximately equals the approach. If the range is extended much further the size of the tower selected will be unduly large and its cost correspondingly greater. Selecting at ranges that are much less than the approach, as with an undersized tower will require that more water be pumped through the condenser, thus increasing the pumping cost. In these 80° wet bulb areas this indicates that a tower should be selected that has a range and an approach of about 7½°, or that will give water off the tower at about 87° or 88°. Now if the water must enter the tower at 95° this means that it will pick up 7° or 8° going through the condenser.

The heat that must be removed from the condenser of the Lennox three ton unit is about 47,500 Btuh. This includes the heat removed from the room plus the heat added by the compressors. To determine the gpm of cooling water that must be circulated divide this total heat by the quantity (500 x Td). The term "Td" is the temperature difference between water entering and water leaving the condenser. In the case mentioned above, where design wet bulb is 80° and range equals approach, "Td" would be 7½°. Using this value for "Td" gives an answer of about 12.5 gpm of water that must be circulated. For selecting cooling towers this required circulation is represented in gpm per ton

and, in this instance, the answer would be about 4 gpm per ton.

From a summary of the information given above the specific requirements covering a cooling tower for the Lennox 3-ton unit at 80° design wet bulb are one that will cool 4 gpm of water per ton of refrigeration from an entering temperature of 95° to a leaving temperature of 87½° at a design wet bulb of 80° and have a capacity of at least 47,000 Btuh. The practice of selecting cooling towers on a tonnage basis only is quite prevalent in some localities but it frequently leads to trouble with undersized towers in areas where the design wet bulb is high. In localities where the design wet bulb temperature is lower than 80° the procedure outlined above will sometimes result in the selection of one size smaller tower. Whether the saving in initial cost actually justifies using a smaller tower is somewhat questionable, because this saving is usually small from one size to another in these smaller towers.

Examples of three actual towers that might be selected in the 80° design wet bulb condition described are the Marley 102W natural draft tower, the Marley Model 33 induced draft tower and for corrosive atmospheric and water conditions the Coats A-12 forced draft tower.

Waste-Water Installations.

In areas where the cost of operation does not prohibit it, there is no objection to using water directly from the house service and wasting it to the drain after it has passed through the condenser. In planning such an installation anticipate the approximate water requirements. This is easy to do by the same method used for figuring the water quantity through the condenser on a tower installation. For the Lennox three ton unit divide 47,500 Btuh by 500 times the Td. "Td" in this instance will be 95° F. minus the temperature of the water available at the tap. Be extremely careful in determining this water temperature at the tap because it varies according to season, distance from the source of supply, depth of water mains and rate of usage in the vicinity. An actual temperature measurement at the location during the months of July and August is quite dependable; during other months it is very unreliable. The local water department is a good place to get information on what maximum water temperature to plan for and the maximum temperature must be used in figuring the water requirement.

In order to maintain sufficient head pressure under light load conditions when wasting water, a flow valve must be used in the water supply to each condenser. The flow valve is a device for regulating the quantity of water passing through the condenser according to the compressor discharge, or head pressure. The higher the head pressure the more water is permitted to flow.

Selecting a Circulating Pump For Tower Applications.

In areas where the design wet bulb is 80°, the Lennox three ton air conditioning unit requires about 12.5 gpm of water circulation with a cooling tower. In areas where design wet bulb is lower the required rate of circulation will be less, but in these small capacities, the size of the pump will ordinarily not be affected. For any climate it is advisable to select a pump, for this unit, on the basis of at least 12 gpm.

Pumps are rated according to the gpm of water they will pump against a given pressure head measured in feet of water. One foot of water head is the pressure exerted by a column of water one foot high. It is customary to measure the pressure of air distribution systems in tenths of inches of water and these water circulating systems are very similar in that their pressure is measured in feet of water. For those who are used to thinking of water pressure as measured in pounds per square inch, a pressure of one lb. per sq. in. is equivalent to about 2.3 feet of water head, or 1 foot of water head corresponds to a pressure of 0.433 lbs. per sq. in.

Head Loss In the Piping.

At a flow rate of 12 gpm the following head losses due to friction can be expected with different sizes of horizontal pipe:

Pipe Size	Loss in Feet of Head per 100 ft. of Pipe
1"	16.4 feet
1¼"	4.3 feet
1½"	2.01 feet
2"	0.79 feet

This table of values serves to illustrate the immense change in pumping head sometimes resulting from a change of only one denomination in pipe size. It demonstrates further that oversizing of pipe assists very little in reducing pumping head unless the system of piping is quite extensive.

To actually size the piping and select a pump begin by making a line sketch of the water system between the conditioning unit, tower and pump, showing, as nearly as possible, all elbows, special fittings and lengths of pipe. If the system of piping is reasonably short (50 feet or less from condensing unit to tower) select a pipe size from the Friction Loss Tables that will give a loss of about 4 ft. per 100 ft. of pipe. If the piping system is long (more than 50 ft. from condensing unit to tower) select a pipe size for a drop of 2 to 3 ft. per 100 ft. of pipe. In the case of a circulating water quantity of 12 gpm this would be 1¼" pipe for the short system and 1½" pipe for the long system.

With the pipe size preliminarily established select from a chart giving resistances of fittings the equivalent lengths of all fittings, and, to this, add the expected length of straight pipe. Multiplying this total equivalent length of pipe by the head loss per 100 ft. and dividing by 100 gives the total head loss of the piping system.

Other Head Losses.

To this head loss of the piping system must be added the drop through the condensing unit for the flow rate selected; also the pumping head of the tower. In reading the cooling equipment performance chart on pressure drop through the condenser for various water rates remember that the two cooling units are always piped in parallel. The

pressure drop from the performance chart is the figure to use in obtaining total pumping head. Do not multiply this chart value by two because there are two units. The pressure drop obtained from this chart will be in pounds per square inch; to convert this to feet of pumping head multiply by 2.3. The pumping head of the tower should be obtained from the tower manufacturer's literature. At 12 gpm this tower pumping head usually is about 20 to 25 feet for a natural draft tower and only about 4 to 6 feet for an induced or forced draft tower.

Use Total Pumping Head and GPM To Select Pump.

The total of piping system head loss, drop through the condensing unit and tower pumping head, is the resistance, expressed in feet of water column, against which the pump must supply the necessary rate of water circulation. With this total pumping head and the gpm of circulation required the proper pump can be selected from any reliable pump manufacturer's published tables. If the manufacturer's tables indicate that the pump required could be replaced by one that is one size smaller, were the total pumping head just a few feet less and, if the difference in price between these two sizes of pumps is substantial, it may be advisable to select one size larger pipe, refigure the pumping head and compare the reduced cost of the pump with the increased cost of the pipe. This is assuming that increasing the pipe size by one denomination results in a total head that will permit selecting one size smaller pump. A little experience in selecting pipe sizes and pumps gives the judgment necessary to determine whether or not it is worthwhile to refigure on the basis of larger pipe.

Pump and Piping Installation.

Of paramount importance to the operation of the system is the actual location of the pump on the job. Without exception the pump casing should be the lowest point in the system. Located high it will tend to lose its prime causing endless service trouble. Its suction should be taken directly from the sump of the tower with as few intervening fittings as possible. The only essential fitting in the pump suction is the strainer in the tower sump. If any valves are used in the pump suction they should be gate or plug-cock valves, never globe valves. Globe valves should never be used in the main water circulating system because of their exteremely high resistance to flow. The only place they may be used is in a condenser bypass which will be explained later.

Another reason for locating the pump casing at the low point of the system is to permit proper drainage in the winter. There is always a removable plug in the bottom of the pump case and the system should be piped in such a manner that every bit of water it holds will drain through this plug. Draining in this manner prevents damage by freezing. In extreme southern climates a stop-and-waste is sometimes used in the pump suction line with the intent that it will hold the water in the tower sump while permitting drainage of the rest of the system through the pump casing. The purpose of this is to hold water in the tower sump all winter to prevent drying and shrinkage of the sump boards which causes excessive leakage when the system is put into operation in the spring. In the extreme south this may be done but where freezing occurs to any extent it will damage the system unless everything is completely drained.

An additional fitting in the water circulating system that is very desirable, though it increases pumping head is a "Y-Pattern" basket strainer. It is particularly helpful in preventing clogged nozzles with an atmospheric tower installation and it assists in keeping the condenser passes clean. It may be placed anywhere in the system, except in the pump suction.

If its drive motor is totally enclosed, as most of them are, the pump may be located anywhere. There is even some advantage to locating it outdoors with the tower if electrical service does not present too much of a problem. Locating it near the tower advantageously shortens its suction line and removes another source of noise from the house. Though not essential it is desirable to provide a sheet metal or wooden box cover for the pump, mainly to prevent tampering, when it is located outdoors.

Where a cooling tower is used it is common practice to install a connection, provided with a globe valve between the cooling water inlet and outlet of the condenser. This alternate path for the cooling water is called a "Condenser Bypass". Its purpose is to provide a means, by adjusting the globe valve, of balancing and adjusting the flow of water through the condenser. This adjustment of water quantity through the condenser is desirable because it is very seldom that the selection of a pump results in one whose characteristics, together with those of the piping system result in exactly the desired flow of water. If a condenser bypass were not used this water flow through the condenser would in almost every case be more than required because the pump is generally selected with a little reserve capacity. With the condenser bypass it is possible to adjust the water rate through the condenser to the desired quantity by measuring the condenser outlet water temperature and setting the bypass valve until the desired water temperature results. The recommended condenser outlet water temperature for the Lennox air conditioning unit is about 95° F. It is important, particularly with a capillary tube system that this water temperature leaving the condenser be adjusted as nearly as possible to 95°. Lower condenser outlet temperature results in less refrigerant flow and reduced cooling capacity. Higher condenser outlet temperature results in increased back pressure and poor latent control.

Section VII—Water Treatment.

Another advantage in selecting a pump with some reserve capacity and providing a bypass is that it permits adjustment to compensate for moderate scaling of the condenser.

Almost all localities have varying quantities and types of minerals in their water supply. Many of

these minerals cause serious problems with water cooled air conditioning equipment because of their tendency to form hard scale in the water passes of the condenser. The condenser is the focal point of this scaling trouble because it is the point of highest temperature in the system. The effect of this scaling is twofold; it acts as an insulator of the condenser tubes reducing the rate of heat transfer between refrigerant and water and it restricts the water passes of the condenser reducing the rate of water flow. The symptoms of scaling are higher than normal compressor head pressure and reduced system capacity. In some localities the water can be so hard as to cause these scaling symptoms in just one season of operation.

With cooling tower applications scaling presents more of a problem than where water is wasted. The reason is that the cooling tower acts as a concentrator of the mineral salts in the water. The process of evaporation that brings about the cooling of the water in the tower removes water constantly but it does not remove the minerals that are being added continually by the makeup water. The concentration of these minerals continues to build up in the circulating water unless corrective measures are taken.

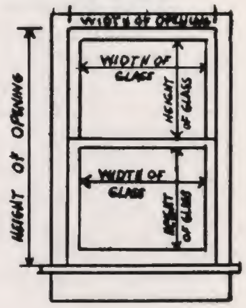
One good means of reducing the rate of condenser liming by reducing this buildup of mineral concentration is to bleed water in small quantities from the cooling tower sump. This rate of "bleedoff" as it is called ought to be about 6 gallons per hour for a three ton installation and most towers are provided with a connection in the sump for taking bleedoff; frequently it is combined with a sump overflow fitting.

The bleedoff, by itself, is usually not sufficient to eliminate troubles with condenser liming. In most cases it is necessary to use some form of water treatment also. The nature of this water treatment varies from one locality to another according to the nature of the hardness of the water supply. The treatment that is used should naturally be of a type that is not harmful to the copper, brass and steel components of the water system and, before it is prescribed, its effectiveness with the water available should be checked either with the local water department or with a competent Water Treatment Consultant.

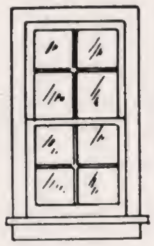
TABLE I — Heat Loss Area in Square Feet, Two Light, Double-Hung Windows, Including Sash

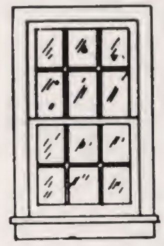
Height of Glass in.	Height of Opening ft.—in.	16	20	24	28	32	36	40	44	48	52
		Width of Opening—Feet-Inches									
		1′-8	2′-0	2′-4	2′-8	3′-0	3′-4	3′-8	4′-0	4′-4	4′-8
20	3′-10	6.4	7.7	8.9	10.2	11.5	12.8	14.0	15.3	16.6	17.9
24	4′-6	7.5	9.0	10.5	12.0	13.5	15.0	16.5	18.0	19.5	21.0
28	5′-2	8.6	10.3	12.0	13.8	15.5	17.2	18.9	20.7	22.4	24.0
32	5′-10	9.7	11.7	13.6	15.5	17.5	19.5	21.4	23.4	25.3	27.2
36	6′-6	10.8	13.0	15.1	17.3	19.5	21.6	23.8	26.0	28.1	30.3
40	7′-2	11.9	14.3	16.7	19.1	21.5	23.8	26.2	28.6	31.0	33.4
44	7′-10	13.0	15.6	18.2	20.8	23.5	26.1	28.7	31.3	34.0	36.5

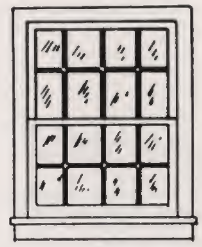
(Column heads 16–52: Width of Glass—Inches)



Heat Area in Square Feet, Multiple Light, Double Hung Windows, Including Sash

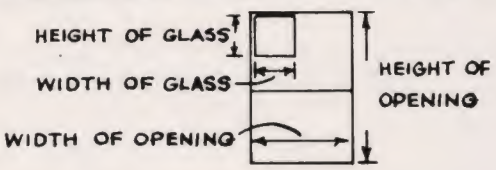






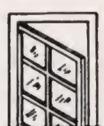
Height of Glass in.	Height of Opening ft.-in.	4-Light Width of Glass—Inches 10	12	14	8-Light Width of Glass—Inches 8	9	10	12	12-Light Width of Glass—Inches 8	9	10	16-Light Width of Glass—Inches 8	9	10
		2′-1″	2′-5″	2′9″	1′-9″	1′-11″	2′-1″	2′-5″	2′-5″	2′-8″	2′-11″	3′-0″	3′-4″	3′-8″
10	3′-10″				6.7	7.5	8.0	9.2	9.2	10.2	11.3	11.5	12.8	14.1
12	4′- 6″				7.9	8.8	9.4	10.9	10.9	12.0	13.3	13.5	15.0	16.5
13	4′-10″											14.5	16.1	17.7
14	5′-2″				9.0	10.1	10.8	12.5	12.5	13.8	15.3	15.5	17.2	19.0
20	3′-10″	8.0	9.2	10.5										
22	4′- 2″	8.7	10.1	11.5										
24	4′- 6″	9.4	10.9	14.4										
26	4′-10″	10.1	11.7	13.3										
28	5′- 2″	10.8	12.5	14.2										
30	5′- 6″	11.5	13.3	15.1										
32	5′-10″	12.2	14.1	16.0										

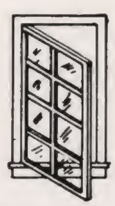
(Second header row: Width of Opening—Feet-Inches)

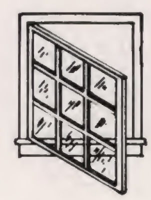


EXAMPLE: 12 LIGHT 9″×12″ WINDOW HAS AN AREA OF 2 SQUARE FEET

Heat Loss Area in Square Feet, Casement Windows, Including Sash







Height of Glass in.	Height of Opening ft.-in.	6-Light Width of Glass—Inches 8	9	10	8-Light Width of Glass—Inches 8	9	10	9-Light Width of Glass—Inches 8	9	10
		1′-8¼″	1′-10¼″	2′-¼″	1′-8¼″	1′-10¼″	2′-¼″	2′-4″	2′-7″	2′-10″
10	2′-11½″	5.0	5.5	6.0				6.9	7.6	8.4
10	3′- 9¾″				6.4	7.1	7.7			
12	3′- 5½″	5.9	6.4	6.9				8.1	8.9	9.8
12	4′- 5¾″				7.6	8.3	9.7			
14	3′-11½″	6.7	7.3	8.0						
14	5′- 1¾″				8.7	9.5	10.4			

(Second header row: Width of Opening—Feet-Inches)

TABLE II

Total Sun and Transmission Factors For Glass Block and Windows.

lj Glass block	18	66	137	93	24	17	21	17	
li Single glass — outside shading screen	25	59	76	63	27	24	24	24	58
lh Single glass — outside venetian blind	27	79	107	87	31	26	26	26	78
lg Single glass — inside venetian blind	31	127	178	140	38	30	30	30	124
lf Single glass — inside roller shade	32	134	191	148	39	31	31	31	131
le Single glass — canvas awning	26	72	96	79	30	25	25	25	71
ld Single Glass — unshaded	35	162	229	179	43	33	33	33	158
Glass Facing	**N**	**NW**	**W**	**SW**	**S**	**SE**	**E**	**NE**	**HORIZ**

NOTES:

1. For thermopane multiply the above table values by 0.85.
2. Figure doors as windows.
3. Where greater than a 20° design T. D. must be used, add 1.0 to above table values for single glass, and add 0.5 for glass block for each degree that exceeds the 20° T. D. Where less than a 20° design T. D. is used, deduct the corresponding quantities.

TABLE III

Total Equivalent Temperature Differentials For Calculating Heat Gain Through Sunlit and Shaded Walls.

North Latitude Wall Facing	4 P.M. Exterior color of Wall—D=dark, L=light		South Latitude Wall Facing
	D	L	
	Frame		
NE	14	14	SE
E	14	14	E
SE	16	14	NE
S	26	20	N
SW	40	28	NW
W	40	28	W
NW	24	20	SW
N (Shade)	14	14	S (Shade)
	4 In. Brick or Stone Veneer + Frame		
NE	12	10	SE
E	12	12	E
SE	18	14	NE
S	26	18	N
SW	32	22	NW
W	26	18	W
NW	12	12	SW
N (Shade)	10	10	S (Shade)
	8 In. Hollow Tile or 8 In. Cinder Block		
NE	10	6	SE
E	20	12	E
SE	20	14	NE
S	24	14	N
SW	12	10	NW
W	10	8	W
NW	8	6	SW
N (Shade)	6	6	S (Shade)
	8 In. Brick or 12 In. Hollow Tile or 12 In. Cinder Block		
NE	14	8	SE
E	18	10	E
SE	18	12	NE
S	10	6	N
SW	10	6	NW
W	10	6	W
NW	6	4	SW
N (Shade	2	2	S (Shade)
	12 In. Brick		
NE	10	4	SE
E	12	8	E
SE	10	6	NE
S	6	4	N
SW	10	6	NW
W	10	6	W
NW	8	4	SW
N (Shade)	2	2	S (Shade)
	8 In. Concrete or Stone or 6 In. or 8 In. Concrete Block		
NE	10	6	SE
E	18	10	E
SE	18	12	NE
S	16	12	N
SW	14	10	NW
W	12	8	W
NW	6	6	SW
N (Shade)	4	4	S (Shade)
	12 In. Concrete or Stone		
NE	14	8	SE
E	18	12	E
SE	16	10	NE
S	10	6	N
SW	8	6	NW
W	10	6	W
NW	6	4	SW
N (Shade)	2	2	S (Shade)

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TABLE IIIA. Total Equivalent Temperature Differentials for Calculating Heat Gain Through Sunlit and Shaded Roofs

Light Construction Roofs—Exposed To Sun	
1″ Wood or 1″ Wood + 1″ or 2″ Insulation	50
Medium Construction Roofs—Exposed To Sun	
2″ Concrete or 2″ Concrete + 1″ or 2″ Insulation or 2″ Wood	50
2″ Gypsum or 2″ Gypsum + 1″ Insulation 1″ Wood or 2″ Wood or 2″ Concrete or 2″ Gypsum } + 4″ Rock Wool in Furred Ceiling	54
4″ Concrete or 4″ Concrete with 2″ Insulation	52
Heavy Construction Roofs—Exposed To Sun	
6″ Concrete	46
6″ Concrete + 2″ Insulation	42
Roofs Covered With Water—Exposed To Sun	
Light Construction Roof with 1″ Water	18
Heavy Construction Roof with 1″ Water	14
Any Roof with 6″ Water	10
Roofs With Roof Sprays—Exposed To Sun	
Light Construction	16
Heavy Construction	12
Roofs In Shade	
Light Construction	14
Medium Construction	12
Heavy Construction	8

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TABLE IV—Heat Gain Factors for Common Building Construction

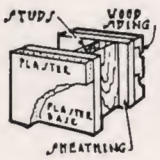
	DESCRIPTION	Heat Transmission Coefficient U	HEAT GAIN FACTOR (Multiply values shown by exposed areas in sq. ft.)									
			Design Temperature Difference, deg. F.									
			4°	8°	12°	16°	20°	24°	28°	32°	36°	40°
	**WINDOWS** **No. 1**											
	(a) Glass, Single	1.13	5	9	14	18	23	27	32	36	41	45
	(b) Glass, double (storm sash)—tight fitting	0.45	2	4	5	7	9	11	13	14	16	18
Window	(c) Storm sash put up and taken down annually will probaby be loose fitting. Under such conditions, recommend using	0.75	3	6	9	12	15	18	21	24	27	30
	**DOORS** **No. 2**											
	(a) Doors are figured the same as though they were windows (See No. 1)											
	EXPOSED WALLS **No. 3 Frame**											
	(a) Frame, wood siding, paper, sheathing, lath and plaster	0.25	1.0	2.0	3.0	4	5	6	7	8	9	10
	(b) Same as (3a) substituting ½" rigid insulation for lath	0.19	.8	2.0	2.0	3	4	5	5	6	7	8
	(c) Same as (3a) with ½" flexible insulation between studs in contact with sheathing	0.17	.7	1.0	2	3	3	4	5	5	6	7
	(d) Same as (3c) with two air spaces	0.15	.6	1.0	2	2	3	4	4	5	5	6
	(e) Same as (3a) with 2" blanket or bat insulation between studs	0.12	.5	1.0	1	2	2	3	3	4	4	5
	(f) Same as (3a) with 3⅝" mineral wool or equivalent between studs	0.09	.4	.7	1	1	2	2	3	3	3	4
	(g) Same as (3a) substituting 25/32" rigid insulation for wood sheathing	0.19	.8	2.0	2	3	4	5	5	6	7	8
	(h) Same as (3a) with composition siding over wood siding	0.21	.8	2	3	3	4	5	6	7	8	8
	(i) Same as (3a) substituting asphalt or asbestos shingles for wood siding	0.30	1.0	2	4	5	6	7	8	10	11	12
Frame	(j) Same as (3h) substituting ½" rigid insulation for lath	0.22	1.0	2	3	4	4	5	6	7	8	9
	(k) Same as (3h) with ½" flexible insulation between studs in contact with sheathing	0.19	.8	2	2	3	4	5	5	6	7	8
	(l) Same as (3j) with two air spaces	0.17	.7	1	2	3	3	4	5	5	6	7
	(m) Same as (3h) with 2" blanket or bat insulation between studs	0.12	.5	1	1	2	2	3	3	4	4	5
	(n) Same as (3h) with 3⅝" mineral wool or equivalent between studs	0.09	.4	1	1	1	2	2	3	3	3	4
	No. 4 Leaky Frame											
Leaky Frame	(a) Clapboards or wood siding, studs, lath and plaster	1.00	4	8	12	16	20	24	28	32	36	40
	(b) Same as (4a) with composition siding over wood siding	0.28	1	2	3	5	6	7	8	9	10	11
	(c) Corrugated sheet metal siding on studs	2.00	8	16	24	32	40	48	56	64	72	80

TABLE IV—Heat Gain Factors for Common Building Construction—Continued

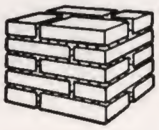
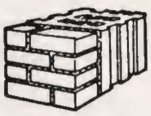
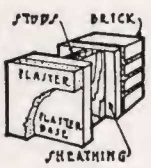
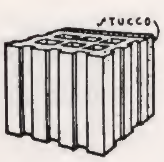
	DESCRIPTION	U	HEAT GAIN FACTOR (Multiply values shown by exposed area in sq. ft.) Design Temperature Difference, deg. F. 4°	8°	12°	16°	20°	24°	28°	32°	36°	40°
 Brick	**No. 5 Brick**											
	(a) 8" Brick wall-plain	0.50	2	4	6	8	10	12	14	16	18	20
	(b) Same as (5a) plastered on one side	0.46	2	4	6	7	9	11	13	15	17	18
	(c) Same as (5a) furred, lath and plaster one side	0.30	1	2	4	5	6	7	8	10	11	12
	(d) Same as (5a) substituting ½" rigid insulation for the lath	0.22	1	2	3	4	4	5	6	7	8	9
	No. 6 Brick											
	(a) 12" Brick wall—plain	0.36	1	3	4	6	7	9	10	12	13	14
	(b) Same as (6a) plastered on one side	0.34	1	3	4	5	7	8	10	11	12	14
	(c) Same as (6a) furred, lath and plaster one side	0.24	1	2	3	4	5	6	7	8	9	10
	(d) Same as (6a) substituting ½" rigid insulation	0.19				3	4	5	5	6	7	8
 Brick & Tile	**No. 7 Brick and Tile**											
	(a) 4" Brick and 8" Hollow Tile	0.34	1	3	4	5	7	8	10	11	12	14
	(b) Same as (7a) plastered on one side	0.33	1	3	4	5	7	8	9	11	12	13
	(c) Same as (7a) furred, lath, and plaster	0.24	1	2	3	4	5	6	7	8	9	10
	(d) Same as (7a) substituting ½" rigid insulation for lath	0.18	1	1	2	3	4	4	5	6	6	7
 Brick Veneer	**No. 8 Brick Veneer**											
	(a) Brick veneer—4" brick, paper, wood sheathing, studs, lath and plaster	0.27	1	2	3	4	5	6	8	9	10	11
	(b) Same as (8a) substituting ½" rigid insulation for lath	0.20	1	2	2	3	4	5	6	6	7	8
	(c) Same as (8a) with 2" blanket or bat insulation between studs	0.12	.5	1	1	2	2	3	3	4	4	5
	(d) Same as (8a) with 3⅝" mineral wool or equivalent between studs	0.09	.4	1	1	1	2	2	3	3	3	4
	(e) Same as (8a) substituting 25/32" rigid insulation in place of wood sheathing	0.21	1	2	3	3	4	5	6	7	8	8
 Tile & Stucco	**No. 9 Tile and Stucco**											
	(a) 8" Hollow Tile—plain stucco exterior	0.40	2	3	5	6	8	10	11	13	14	16
	(b) Same as (9a) plastered	0.38	2	3	5	6	8	9	11	12	14	15
	(c) Same as (9a) furred, lath and plaster	0.26	1	2	3	4	5	6	7	8	9	10
	(d) Same as (9c) substituting ½" rigid insulation for lath	0.20	1	2	2	3	4	5	6	6	7	8
	No. 10 Tile and Stucco											
	(a) 12" Hollow Tile—plain stucco exterior	0.30	1	2	4	5	6	7	8	10	11	12
	(b) Same as (10a) plastered	0.29	1	2	3	5	6	7	8	9	10	12
	(c) Same as (10a) furred, lath and plaster	0.22	1	2	3	4	4	5	6	7	8	9
	(d) Same as (10c) substituting ½" rigid insulation for lath	0.17	1	1	2	3	3	4	5	5	6	7

TABLE IV—Heat Gain Factors for Common Building Construction—Continued

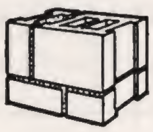
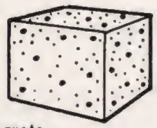
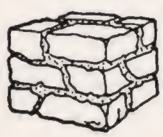
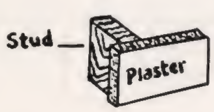
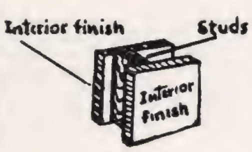
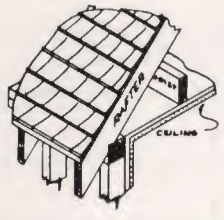
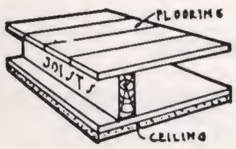
	DESCRIPTION	U	HEAT GAIN FACTOR (Multiply values shown by exposed areas in sq. ft.) Design Temperature Difference, deg. F. 4°	8°	12°	16°	20°	24°	28°	32°	36°	40°
 Blocks	**No. 11 Block**											
	(a) 8″ Cinder Block—plain (Also Haydite)	0.42	2	3	5	7	8	10	12	13	15	17
	(b) Same as (11a) plastered	0.39	2	3	5	6	8	9	11	12	14	16
	(c) Same as (11a) furred, lath and plaster	0.27	1	2	3	4	5	6	8	9	10	11
	(d) Same as ((11c) substituting ½″ rigid insulation for lath	0.20	1	2	2	3	4	5	6	6	7	8
	No. 12 Block											
	(a) 8″ Concrete Block—plain	0.56	2	4	7	9	11	13	16	18	20	22
	(b) Same as (12a) plastered	0.52	2	4	6	8	10	12	15	17	19	21
	(c) Same as (12a) furred, lath and plaster	0.32	1	3	4	5	6	8	9	10	12	13
	(d) Same as (12c) substituting ½″ rigid insulation for lath	0.24	1	2	3	4	5	6	7	8	9	10
	(e) Same as (12a) basement wall, below grade	0.10	.4	1	1	2	2	2	3	3	4	4
	(f) 12″ concrete block, above grade	0.50	2	4	6	8	10	12	14	16	18	20
	(g) 12″ concrete block, below grade	0.10	.4	1	1	2	2	2	3	3	4	4
 Concrete	**No. 13 Concrete**											
	(a) 8″ Concrete wall, above grade	0.70	3	6	8	11	14	17	20	22	25	28
	(b) 8″ concrete wall, below grade	0.10	.4	1	1	2	2	2	3	3	4	4
	(c) 12″ concrete wall, above grade	0.58	2	5	7	9	12	14	16	19	21	23
	(d) 12″ concrete wall, below grade	0.10	.4	1	1	2	2	2	3	3	4	4
 Stone	**No. 14 Stone**											
	(a) 8″ Limestone or Sandstone—plain	0.71	3	6	9	11	14	17	20	23	26	28
	(b) Same as (14a) plastered on one side	0.64	3	5	8	10	13	15	18	20	23	26
	(c) Same as (14a) furred, lath and plaster	0.37	1	3	4	6	7	9	10	12	13	15
	(d) Same as (14c) substituting ½″ rigid insulation for lath	0.25	1	2	3	4	5	6	7	8	9	10
	(e) 12″ limestone, below grade	0.10	.4	1	1	2	2	2	3	3	4	4
	(f) 16″ limestone, below grade	0.10	.4	1	1	2	2	2	3	3	4	4
	No. 15 Stone											
	(a) 12″ Limestone or Sandstone—plain	0.58	2	5	7	9	12	14	16	19	21	23
	(b) Same as (15a) plastered on one side	0.53	2	4	6	8	11	13	15	17	19	21
	(c) Same as (15a) furred, lath and plaster	0.33	1	3	4	5	7	8	9	11	12	13
	(d) Same as (15c) substituting ½″ rigid insulation for lath	0.23	1	2	3	4	5	6	6	7	8	9
Glass 	**No. 16 Glass Block**											
	(a) 3⅞″ Glass Block	0.49	2	4	6	8	10	12	14	16	18	20
	FRAME INTERIOR PARTITIONS											
	**No. 17 Finish One Side**											
	(a) With lath and plaster one side: other side open	0.62	2	5	7	10	12	15	17	20	22	25
	(b) Same as (17a) substituting ½″ rigid insulation for lath	0.35	1	3	4	6	9	8	10	11	13	14
	(c) Same as (17a) with ½″ rigid insulation on opposite side	0.25	1	2	3	4	5	6	7	8	9	10
	**No. 18 Finish Both Sides**											
	(a) With lath and plaster both sides	0.34	1	3	4	5	7	8	10	11	12	14
	(b) Same as (18a) substituting ½″ rigid insulation for lath	0.18	1	1	2	3	4	4	5	6	6	7
	(c) Same as (18a) with 2″ blanket or bat insulation between studs	0.12	.5	1	1	2	2	3	3	4	4	5
	(d) Same as (18a) with 3⅝″ mineral wool insulation or equivalent	0.09	.4	1	1	1	2	2	3	3	3	4

TABLE IV—Heat Gain Factors for Common Building Construction—Continued

HEAT GAIN FACTOR (Multiply values shown by exposed areas in sq. ft.) — Design Temperature Difference, deg. F.

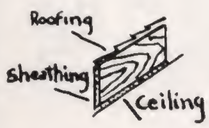


DESCRIPTION	U	30°	32°	34°	36°	38°	40°	42°	44°	46°	48°	50°	52°	54°	56°	58°	60°
CEILINGS WITH ATTIC SPACE ABOVE																	
No. 19 No Floor																	
(a) Lath and plaster, no floor above	0.61	18	20	21	22	23	24	26	27	28	29	31	32	33	34	35	37
(b) Same as (19a) substituting ½" rigid insulation for lath	0.35	11	11	12	13	13	14	15	15	16	17	18	18	19	20	20	21
(c) Same as (19a) with ½" rigid insulation on top of joists	0.25	8	8	9	9	10	10	11	11	12	12	13	13	14	14	15	15
(d) Same as (19a) with 2" blanket or bat insulation between joists	0.12	4	4	4	4	5	5	5	5	6	6	6	6	6	7	7	7
(e) Same as (19a) with 3⅝" mineral wool or equivalent between joists	0.09	3	3	3	3	3	4	4	4	4	4	5	5	5	5	5	5



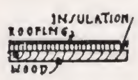
With Roof Above

DESCRIPTION	U	30°	32°	34°	36°	38°	40°	42°	44°	46°	48°	50°	52°	54°	56°	58°	60°
No. 20 With Floor																	
(a) Lath and plaster with tight floor above	0.28	8	9	10	10	11	11	12	12	13	13	14	15	15	16	16	17
(b) Same as (20a) substituting ½" rigid insulation for the lath	0.21	6	7	7	8	8	8	9	9	10	10	11	11	11	12	12	13
(c) Same as (20a) with 2" blanket or bat between joists	0.11	3	4	4	4	4	4	5	5	5	5	6	6	6	6	6	7
(d) Same as (20a) with 3⅝" mineral wool or equivalent between joists	0.09	3	3	3	3	3	4	4	4	4	4	5	5	5	5	5	5



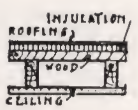
No Attic Space

DESCRIPTION	U	30°	32°	34°	36°	38°	40°	42°	44°	46°	48°	50°	52°	54°	56°	58°	60°
CEILING—PART OF ROOF—NO ATTIC SPACE																	
No. 22																	
(a) Lath and plaster, rafter, sheathing, shingles	0.29	9	9	10	10	11	12	12	13	13	14	15	15	16	16	17	17
(b) Same as (22a) substituting ½" rigid insulation for the lath	0.21	6	6	7	8	8	8	9	9	10	10	11	11	11	12	12	13
(c) Same as (22a) with 2" blanket or bat insulation between joists	0.12	4	4	4	4	5	5	5	5	6	6	6	6	6	7	7	7
(d) Same as (22a) with 3⅞" mineral wool or equivalent between rafters	0.09	3	3	3	3	3	4	4	4	4	4	5	5	5	5	5	5



Flat Roof
No Ceiling

DESCRIPTION	U	30°	32°	34°	36°	38°	40°	42°	44°	46°	48°	50°	52°	54°	56°	58°	60°
FLAT ROOFS WITH BUILT UP ROOFING AND FLAT METAL ROOFS WITH 1" BOARDS																	
No. 23 No Ceiling																	
(a) Wood 1" thick without ceiling, and with no insulation	0.49	15	16	17	18	19	20	21	22	23	24	25	25	26	27	28	29
(b) Wood 1" thick without ceiling, but with ½" rigid insulation (No air space)	0.28	8	9	10	10	11	11	12	12	13	13	14	15	15	16	16	17
(c) Same as (23a) with 2" blanket or bat insulation between joists	0.12	4	4	4	4	5	5	5	5	6	6	6	6	6	7	7	7
(d) Same as (23a) with 3⅝" mineral wool or equivalent between joists	0.09	3	3	3	3	3	4	4	4	4	4	5	5	5	5	5	5
(e) Corrugated sheet metal on rafters	2.00	60	64	68	72	76	80	84	88	93	96	100	104	108	112	116	120

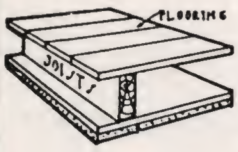


Flat Roof
With Ceiling

DESCRIPTION	U	30°	32°	34°	36°	38°	40°	42°	44°	46°	48°	50°	52°	54°	56°	58°	60°
No. 24 With Ceiling																	
(a) Wood 1" thick with lath and plaster ceiling, and no insulation	0.32	10	10	11	12	12	13	13	14	15	15	16	17	17	18	19	19
(b) Same as (24a) with ½" rigid insulation instead of lath	0.23	7	7	8	8	9	9	10	10	11	11	12	12	12	13	13	14
(c) Wood 1" thick with lath and plaster ceiling, and with ½" rigid insulation	0.21	6	7	7	8	8	8	9	9	10	10	11	11	11	12	12	13
(d) Same as (24a) with 2" blanket or bat insulation between joists	0.12	4	4	4	4	5	5	5	5	6	6	6	6	6	7	7	7
(e) Same as (24a) with 3⅝" mineral wool or equivalent between joists	0.09	3	3	3	3	3	4	4	4	4	4	5	5	5	5	5	5

TABLE IV—Heat Gain Factors for Common Building Construction—Continued

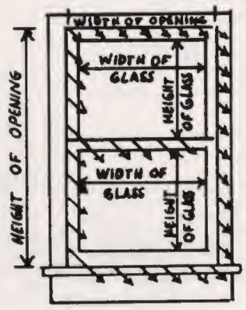
DESCRIPTION	U	HEAT GAIN FACTOR (Multiply values shown by exposed area in sq. ft.) Design Temperature Difference, deg. F. 4°	8°	12°	16°	20°	24°	28°	32°	36°	40°
WOOD FLOORS OVER SPACES EXPOSED TO OUTDOOR TEMPERATURES No. 26											
(a) Double floor on joists	0.34	1	3	4	5	7	8	10	11	12	14
(b) Same as (26a) with lath and plaster ceiling	0.25	1	2	3	4	5	6	7	8	9	10
(c) Same as (26a) with ½" rigid insulation on bottom of joists	0.19	1	2	2	3	4	5	5	6	7	8
(d) Same as (26a) with sheathing on bottom of joists and 2" bat insulation between joists	0.12	.5	1	1	2	2	3	3	4	4	5
(e) Same as (26a) with sheathing on bottom of joists and 3⅝" mineral wool or equivalent between joists	.09	.5	1	1	1	2	2	3	3	3	4



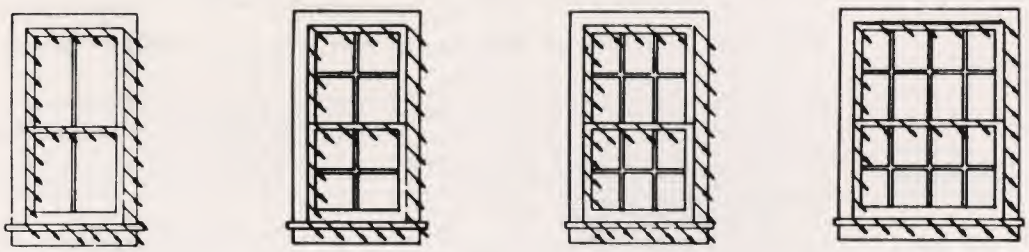
Floor

TABLE V—Running Feet of Crack, Two Light, Double-Hung Windows, Including Sash

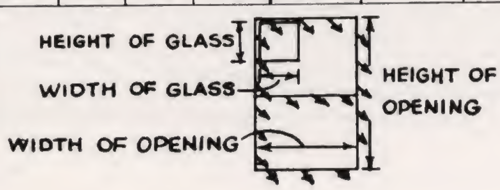
Height of Glass in.	Height of Opening ft.—in.	Width of Glass—Inches 16 / Width of Opening—Feet-Inches 1'-8	20 / 2'-0	24 / 2'-4	28 / 2'-8	32 / 3'-0	36 / 3'-4	40 / 3'-8	44 / 4'-0	48 / 4'-4	52 / 4'-8
20	3'-10	12.7	13.7	14.7	15.7	16.7	17.7	18.7	19.7	20.7	21.7
24	4'-6	14.0	15.0	16.0	17.0	18.0	19.0	20.0	21.0	22.0	23.0
28	5'-2	15.3	16.3	17.3	18.3	19.3	20.3	21.3	22.3	23.3	24.3
32	5'-10	16.7	17.7	18.7	19.7	20.7	21.7	22.7	23.7	24.7	25.7
36	6'-6	18.0	19.0	20.0	21.0	22.0	23.0	24.0	25.0	26.0	27.0
40	7'-2	19.3	20.3	21.3	22.3	23.3	24.3	25.3	26.3	27.3	28.3
44	7'-10	20.7	21.7	22.7	23.7	24.7	25.7	26.7	27.7	28.7	29.7



Running Feet of Crack, Multiple Light, Double-Hung Windows, Including Sash



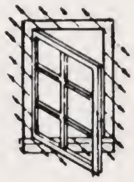
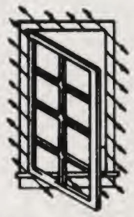
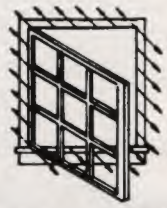
Height of Glass in.	Height of Opening ft.-in.	4-Light 10 / 2'-1"	4-Light 12 / 2'-5"	4-Light 14 / 2'9"	8-Light 8 / 1'-9"	8-Light 9 / 1'-11"	8-Light 10 / 2'-1"	8-Light 12 / 2'-5"	12-Light 8 / 2'-5"	12-Light 9 / 2'-8"	12-Light 10 / 2'-11"	16-Light 8 / 3'-0"	16-Light 9 / 3'-4"	16-Light 10 / 3'-8"
		Width of Glass—Inches / Width of Opening—Feet-Inches												
10	3'-10"				12.9	13.3	13.9	14.9	14.9	15.7	16.4	16.7	17.7	18.7
12	4'- 6"				14.3	14.8	15.3	16.3	16.3	17.0	17.8	18.0	19.0	20.0
13	4'-10"											18.7	19.7	20.7
14	5'-2"				15.6	16.1	16.6	17.6	17.6	18.3	19.1	19.3	20.3	21.3
20	3'-10"	13.9	14.9	15.9										
22	4'- 2"	14.6	15.6	16.6										
24	4'- 6"	15.3	16.3	17.3										
26	4'-10"	15.9	16.9	17.9										
28	5'- 2"	16.6	17.6	18.6										
30	5'- 6"	17.3	18.3	19.3										
32	5'-10"	17.9	18.9	19.9										



EXAMPLE: 12 LIGHT 9"x12" WINDOW HAS A CRACKAGE OF 17.0 FEET.

TABLE V

Running Feet of Crack, Casement Windows, Including Sash

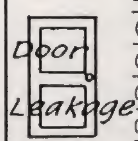




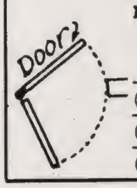
Height of Glass in.	Height of Opening ft.-in.	6-Light Width of Glass—Inches 8 (Width of Opening 1'-8¼")	6-Light 9 (1'-10¼")	6-Light 10 (2'-¼")	8-Light Width of Glass—Inches 8 (Width of Opening 1'-8¼")	8-Light 9 (1'-10¼")	8-Light 10 (2'-¼")	9-Light Width of Glass—Inches 8 (Width of Opening 2'-4")	9-Light 9 (2'-7")	9-Light 10 (2'-10")
10	2'-11½"	9.3	9.6	10.0				10.6	11.0	11.6
10	3'- 9¼"				11.0	11.3	11.7			
12	3'- 5½"	10.3	10.6	11.0				11.6	12.0	12.6
12	4'- 5¾"				13.0	13.3	13.6			
14	3'-11½"	11.3	11.6	12.0						
14	5'- 1¾"				13.7	14.0	14.3			

TABLE VI—Air Leakage—CFM Per Ft. of Crackage.

Window Leakage

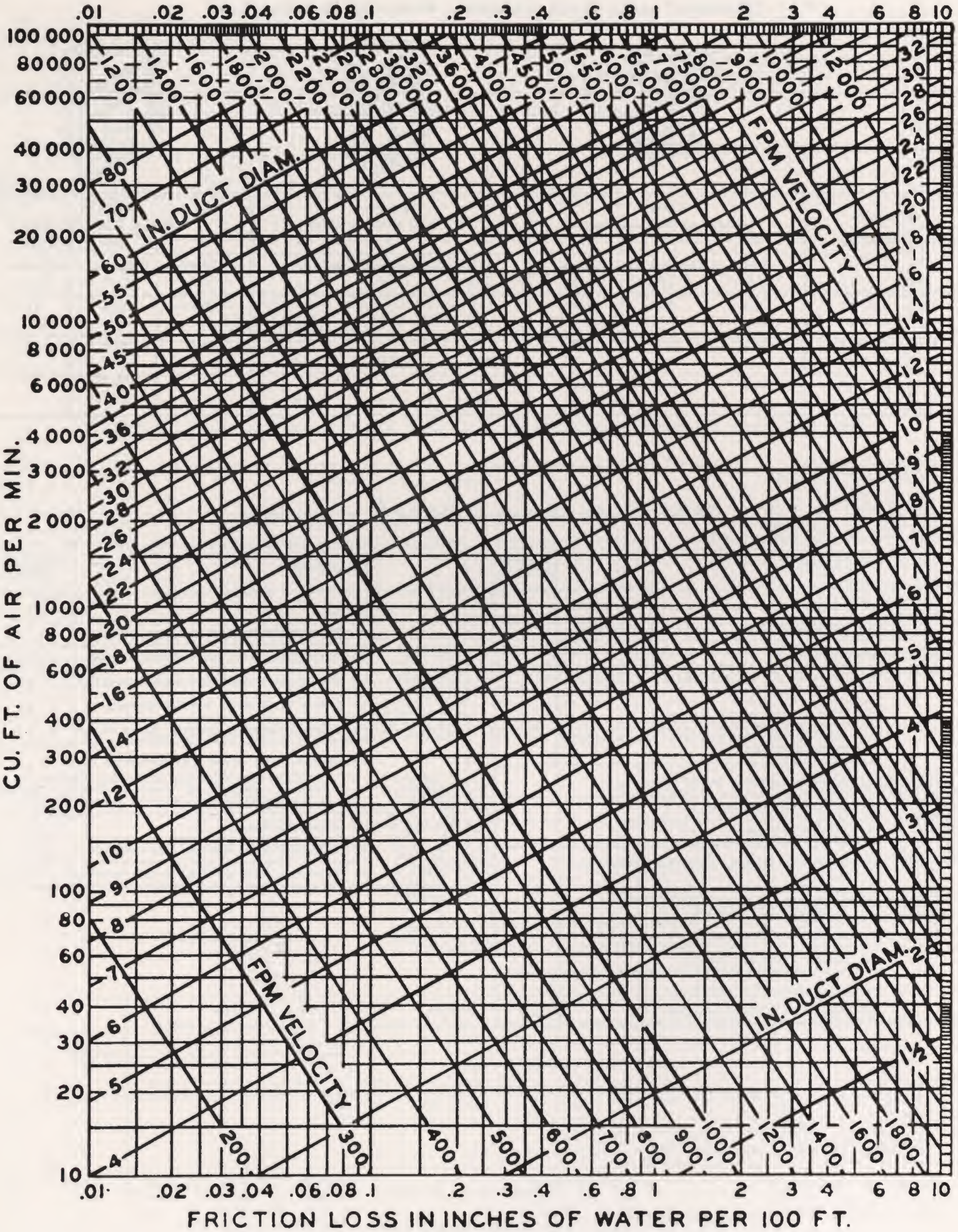
DESCRIPTION	Summer C.F.M./ft.
WINDOWS	
No. 28—Double-Hung, Wood Sash Windows	
(a) Average fit, not weatherstripped	.4
(b) Average fit, weatherstripped or equipped with storm windows	.2
(c) Poor fit, not weatherstripped	1.0
(d) Poor fit, weatherstripped or equipped with storm windows	.4
No. 29—Double-Hung, Metal Windows	
(a) Not weatherstripped	.8
(b) Weatherstripped	.4
No. 30—Rolled Section, Steel Sash Windows	
(a) Industrial pivoted	2.0
(b) Architectural projected	1.0
(c) Residential Casement	.5
(d) Heavy casement section	.4
No. 31—Hollow Metal Windows	
(a) Vertically pivoted window	2.0





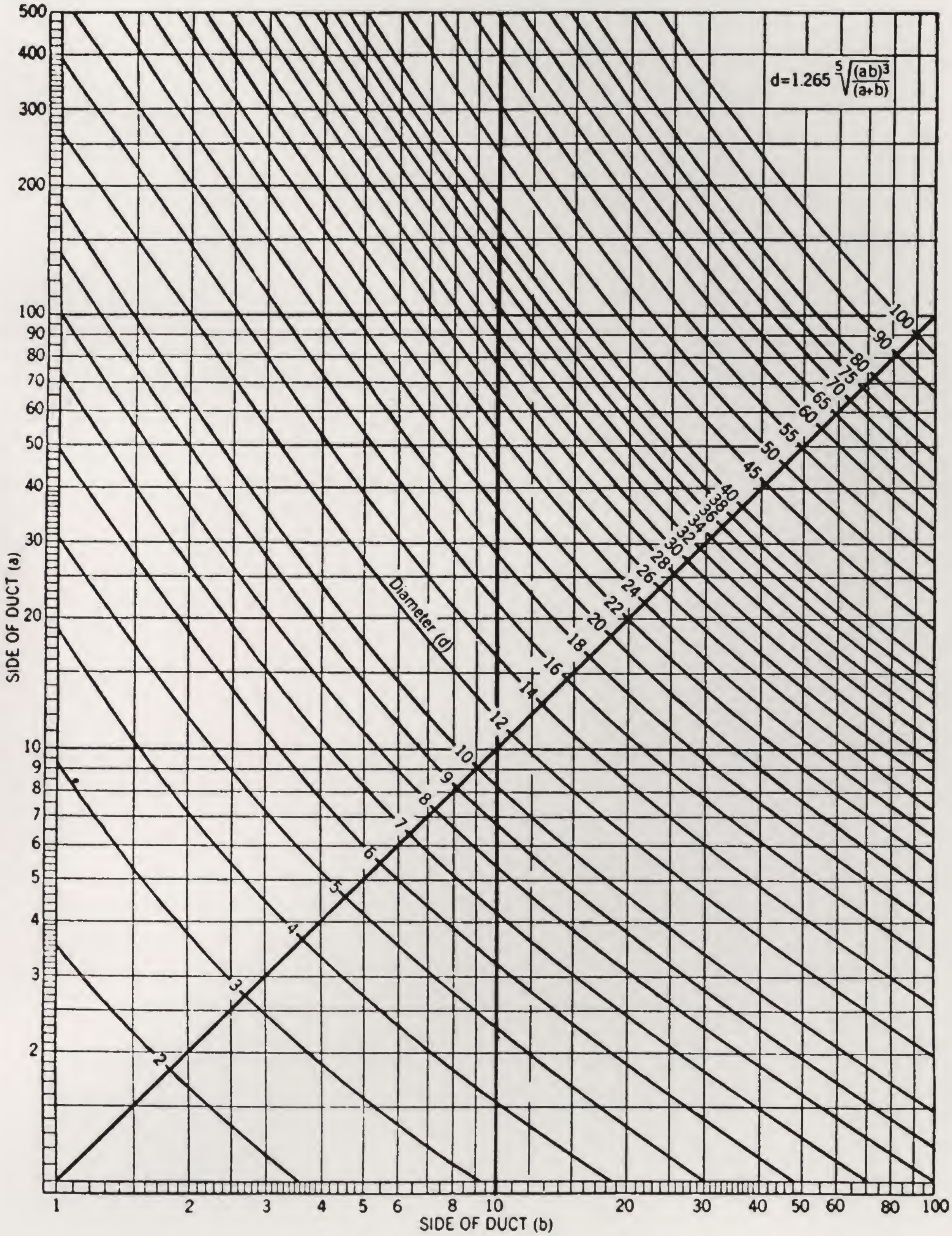
DESCRIPTION	Summer C.F.M./ft.
DOORS	
No. 32—Residential and average service	
(a) Well fitted door, not weatherstripped	1.0
(b) Poorly fitted door, not weatherstripped	2.0
(c) Poorly fitted door, weatherstripped	1.0
(d) Doors frequently opened, as in stores	3.0
No. 33—Swinging Doors and Revolving Doors (multiply the following Infiltration Factors by the average number of persons entering building during period of one hour in severe weather.)	
(a) Swinging doors	2.0
(b) Swinging doors leading into vestibule	1.0
(c) Revolving doors (if used in cold weather)	1.0

TABLE VII



The "Equal Friction" Chart

TABLE VIII



Rectangular Equivalents of Round Ducts

Showing the rectangular ducts that offer the same resistance to air travel per 100 ft. as various sizes of round ducts.